INVESTIGATION OF GAS ENGINE DRIVEN HEAT PUMP SYSTEMS

by

CLIVE HICKMAN

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To Judy

For Yesterday's Memories

Today's Love

And Tomorrow's Dreams

THE UNIVERSITY OF ASTON

INVESTIGATION OF GAS ENGINE DRIVEN HEAT PUMP SYSTEMS

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Summary

Heat pumps are becoming increasingly popular, but poor electricity generating efficiency limits the potential energy savings of electrically powered units. Thus the work reported in this thesis concerns the development of a range of gas engine driven heat pumps for industrial and commercial heating applications, which recover heat from the prime mover, normally rejected to waste.

Despite the convenience of using proprietary engine heat recovery packages, investigations have highlighted the necessity to ensure the engine and the heat recovery equipment are compatible.

A problem common to all air source heat pumps is the formation of frost on the evaporator, which must be removed periodically, with the expenditure of energy, to ensure the continued operation of the plant. An original fluidised bed defrosting mechanism is proposed, which prevents the build up of this frost, and also improves system performance.

Criticisms have been levelled against the rotary sliding vane compressor, in particular the effects of lubrication, which is essential. This thesis compares the rotary sliding vane compressor with other machines, and concludes that many of these criticisms are unfounded.

A confidential market survey indicates an increasing demand for heat pumps up to and including 1990, and the technical support needed to penetrate this market is presented. Such support includes the development of a range of modular gas engine driven heat pumps, and a computer aided design for the selection of the optimum units.

A case study of a gas engine driven heat pump for a swimming pool application which provided valuable experience is included.

KEY WORDS: HEAT PUMPS, GAS ENGINES, FLUIDISED BEDS, REFRIGERANT COMPRESSORS, EVAPORATOR FROSTING

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NOMENCLATURE

A_a	Air side surface area	(m^2)
A_B	Annual running costs of boiler plant	(£/yr)
A_E	Evaporator surface area	(m^2)
A_{EC}	Evaporator surface forced convection	(m^2)
A_{EF}	Evaporator surface fluidised bed	(m^2)
A_H	Annual running cost of heat pump	(£/yr)
A_o	Orifice plate cross sectional area	(m^2)
A_R	Fin base area per unit area basic surface	(dimensionless)
A_r	Refrigerant side surface area	(m^2)
A_s	Fin surface area per unit area basic surface	(dimensionless)
b	Fin thickness	(m)
COP	Coefficient of performance	(dimensionless)
COP_C	Carnot coefficient of performance	(dimensionless)
$\overline{COP}$	Seasonal coefficient of performance	(dimensionless)
Cp	Specific heat capacity	(J/kgK)
Cp_a	Specific heat capacity of air	(J/kgK)
Do	Orifice plate diameter	(m)
do	Orifice plate bore	(m)
d_p	Particle diameter	(m)
E	Fractional engine load factor	(dimensionless)
F	Compressor power temperature compensating factor	(dimensionless)
F_B	Boiler fuel cost	(£/W)
F_E	Electricity fuel cost	(£/W)
F_H	Heat pump fuel cost	(£/W)

G	Modulas of rigidity	(N/m^2)
g	Acceleration due to gravity	(m/s^2)
Ga	Galileo number	(dimensionless)
GEHP	Gas Engine Heat Pump	
H	Ambient enthalpy	(J/kg dry air)
h_a	Specific enthalpy at compressor suction	(J/kg)
h_c	Specific enthalpy at compressor discharge	(J/kg)
h_d	Pressure drop across orifice plate	(mm H_2O)
h_g	Air side heat transfer coefficient	(W/m^2K)
H_i	Settled bed height	(m)
h_i	Specific enthalpy of injected liquid	(J/kg)
h_r	Refrigerant side heat transfer coefficient	(W/m^2K)
I_B	Installed cost of boiler	(£)
I_H	Installed cost of heat pump	(£)
J	Section modulas	(m^4)
k	Heat exchanger thermal conductivity	(W/mK)
l	Length of shaft	(m)
L_B	Boiler life expectancy	(year)
L_H	Heat pump life expectancy	(year)
M_B	Annual maintenance cost - boiler	(£/yr)
M_H	Annual maintenance cost - heat pump	(£/yr)
mE	Dimensionless variable defined in BS 1042	(dimensionless)
$\dot{m}_A$	Mass flow rate at compressor suction	(kg/s)
$\dot{m}_c$	Mass flow rate at compressor discharge	(kg/s)
$\dot{m}_g$	Mass flow rate fluidising gas	(kg/s)
$\dot{m}_i$	Mass flow rate of injected liquid	(kg/s)
N	Engine speed	(rev/min)
n	Number of months in heating season	(dimensionless)
N_a	Dimensionless group defined in BS 1042	(dimensionless)

PER	Primary energy ratio	(dimensionless)
$\overline{PER}$	Seasonal primary energy ratio	(dimensionless)
Q	Annual heat requirement	(J)
Q_B	Rated heat output of boiler	(W)
Q_C	Condenser heat rejected	(W)
Q_E	Evaporator heat extraction rate	(W)
Q_F	Fuel consumption	(W)
Q_H	Rated output of heat pump	(W)
Q_{RB}	Rated power consumption of boiler plant fans/pumps	(W)
Q_{RH}	Rated power consumption of heat pump fans/pumps	(W)
Q_T	Total useful heat rejected by GEHP	(W)
r	Radius of shaft	(m)
Re_{mf}	Reynolds number for incipient fluidisation	(dimensionless)
T	Torque	(Nm)
t	Time	(s)
T_c	Condensing temperature	(^{o}C)
T_e	Evaporating temperature	(^{o}C)
T_H	Temperature of heat sink	(^{o}C)
T_L	Temperature of heat source	(^{o}C)
U	Overall heat transfer coefficient	(W/m^{2}K)
U_c	Overall heat transfer coefficient finned evaporator	(W/m^{2}K)
U_f	Overall heat transfer coefficient fluidised bed	(W/m^{2}K)

$\dot{u}_a$	Volumetric air flow rate	(m^3/s)
$\dot{u}_{ac}$	Air velocity over finned evaporator	(m/s)
$\dot{u}_{af}$	Air velocity through fluidised bed	(m/s)
W	Mechanical work to compressor	(W)
ΔP	Pressure drop across fluidised bed	(N/m^2)
ΔT	Differential temperature	(K)
ΔT_a	Differential air temperature	(K)
ΔT_m	Log mean temperature difference	(K)
η	Fin efficiency	(dimensionless)
η_B	Boiler efficiency	(dimensionless)
ρ_a	Air density	(kg/m^3)
ρ_g	Gas density	(kg/m^3)
ρ_p	Particle density	(kg/m^3)
μ_a	Air viscosity	(kg/ms)

CHAPTER ONE

INTRODUCTION

1.1 PROJECT ORIGIN

This project was initiated by Denco Air Limited who are market leaders for refrigeration and air conditioning plant, and who have, in conjunction with the Ford Motor Company, developed a 'concept engine package' for heat pump applications. This is based upon the Ford 1.6 litre automotive engine, converted to natural gas operation, and the Denco A.G.R. rotary sliding vane compressor. The aim of the project is to monitor field installations of these Gas Engine Driven Heat Pumps (GEHP). The project was organised in collaboration with the Interdisciplinary Higher Degrees Scheme (IHD) [1] which provides an ideal environment for Industrial/Academic projects of this type. At an early stage it was decided that the evaporator should be studied in depth, and in particular the frosting problem of air heated evaporators.

The project moved along two parallel courses, one to monitor an existing gas engine driven heat pump system, the other to evaluate alternative evaporator defrosting techniques.

The co-ordinating role of IHD was invaluable in providing the required academic support.

1.2 OBJECTIVES

The objectives of the project presented in this thesis were as follows:

i) To monitor an existing Gas Engine Driven Heat Pump system and to propose recommendations for future improvements.

ii) To develop a range of packaged heat pump systems based on the existing module.

iii) To produce a computer design aid for the selection of the optimum module for a given application.

iv) To investigate the performance of air heated evaporators, and to evaluate defrosting techniques.

1.3 PROJECT PROGRAMME

To satisfy the objectives, the programme for the project consisted of the following phases:

a) Literature search and review.

b) Experimental data acquisition and analysis for the prototype gas engine driven heat pump.
From this it was apparent that the following areas required detailed investigation:
   i) Compressor performance.
   ii) Performance of the engine heat recovery equipment.
   iii) Evaporator performance in particular defrosting techniques.

c) Examination of the technical support required to enable market penetration.

The thesis follows this pattern.

CHAPTER TWO

REVIEW OF LITERATURE

HEAT PUMPS

2.1 GENERAL REMARKS

A heat pump and a refrigerator are thermodynamically identical. However, in the case of the heat pump it is the hot reservoir which is of interest (see Figure 2.1) whilst the cold reservoir is important for refrigeration. It can be seen from Figure 2.1 that this cycle is simply a heat engine in reverse.

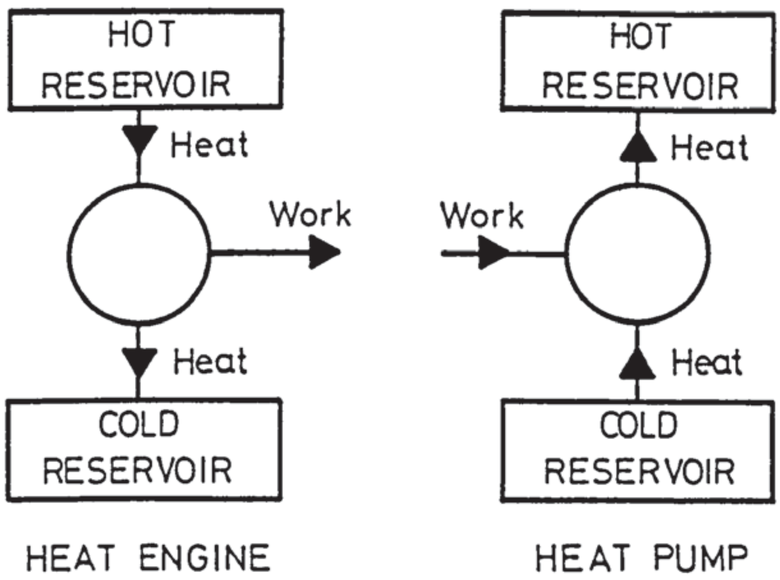


Figure 2.1 Thermodynamic Models Of A Heat Engine And A Heat Pump

Early work on the phenomenon of heat pumping is attributed to W. Thomson [2] who, working on Carnot's 'Cycle of Heat' [3], proposed a method of heating and cooling buildings by means of currents of air as long ago as 1852.

It is interesting to go back further, to the ninth century, as did McMullen et. al. [4], and consider the early works of Paulus Orosius [5] who reports of storing the dead indoors and above ground for several months prior to cremation.

> "There is among the Ests a tribe that can produce cold, and dead men lie there so long without decay because they bring the cold upon them. And if one set down two vats full of ale or water they will arrange for both to be frozen over whether it be summer or winter."

Refrigeration plant based upon the vapour compression cycle has been in use for many years, since it is the only practical method of ice production. Heat pumping on the other hand must compete with alternative direct fired heating systems for which the capital costs are much lower.

Although large energy savings are possible with heat pumps, in an economic environment, it is cost savings rather than energy savings which are of prime importance; consequently until the early 1970s (at the time of the energy crisis and subsequent escalation in fuel prices) heat pump development in the United Kingdom remained inert.

Gas engine driven heat pumps (GEHP) can be advantageous in applications where the waste exhaust and coolant heat (normally rejected via cooling towers during electricity production) can be utilised.

The most commonly used naturally occurring heat source (cold reservoir) is ambient air. However, the high humidity levels prevalent in the United Kingdom create problems due to the frost formation on the face of the heat exchanger (evaporator) in this cold reservoir at ambient temperatures around 0^{o}C. Unless action is taken to remove this frost, heat transfer is suppressed. Heap [6] reports that frost formation and its subsequent removal, results in a loss of between 5% and 10% of the total heat output from the system.

2.2 THERMODYNAMIC PRINCIPLES

There are numerous cycles which can be used for heat pump applications, and these are listed below:

a) Vapour compression cycle

b) Absorption cycle

c) Brayton cycle

d) Stirling cycle

e) Lorenz cycle

In addition thermoelectric heat pumps based on the Peltier effect and chemical heat pumps are also emerging.

The vapour compression cycle is the one applied to most refrigeration plants, and all work reported in this thesis is based upon this cycle. Further, McMullen et. al. [4] suggest that this is the only cycle that is economically viable. The ideal cycle comprises two isentropic (constant entropy) and two isothermal (constant temperature) processes and is shown in Figure 2.2.

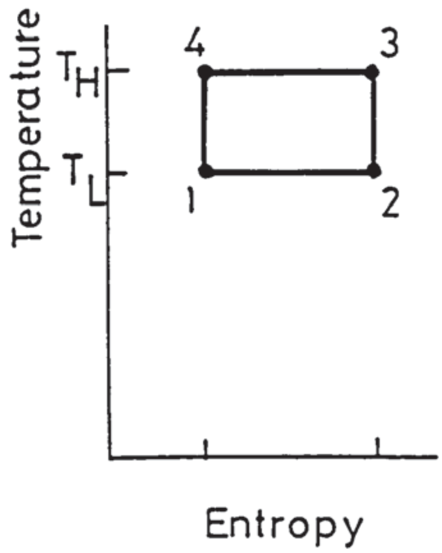


Figure 2.2 The Ideal Carnot Heat Pump Cycle

Heat is received isothermally at T_L, and delivered isothermally at T_H. Isentropic compression, and expansion processes complete the cycle, and energy balance is achieved by work input from an external prime mover.

Heat pump performance is expressed as the ratio Heat Rejected : Work Supplied, and is termed the "Coefficient of Performance" (C.O.P.). From consideration of the temperature - entropy (T-S) diagram, it can be shown that for a Carnot cycle:

$$COP_C = \frac{Q_c}{W} = \frac{T_H}{T_H - T_L} \qquad (2.1)$$

In practice it is impossible to construct this ideal cycle due to numerous limitations. In particular, an expansion valve rather than an expansion machine is used (at the expense of a small amount of useful work), due to cost considerations, and in general dry vapour must be supplied to proprietry compressors, since liquid refrigerant can cause extensive damage and eventual failure.

The practical cycle is shown in Figure 2.3 along with the pressure-enthalpy (p-h) diagram, which is the most convenient for system analysis.

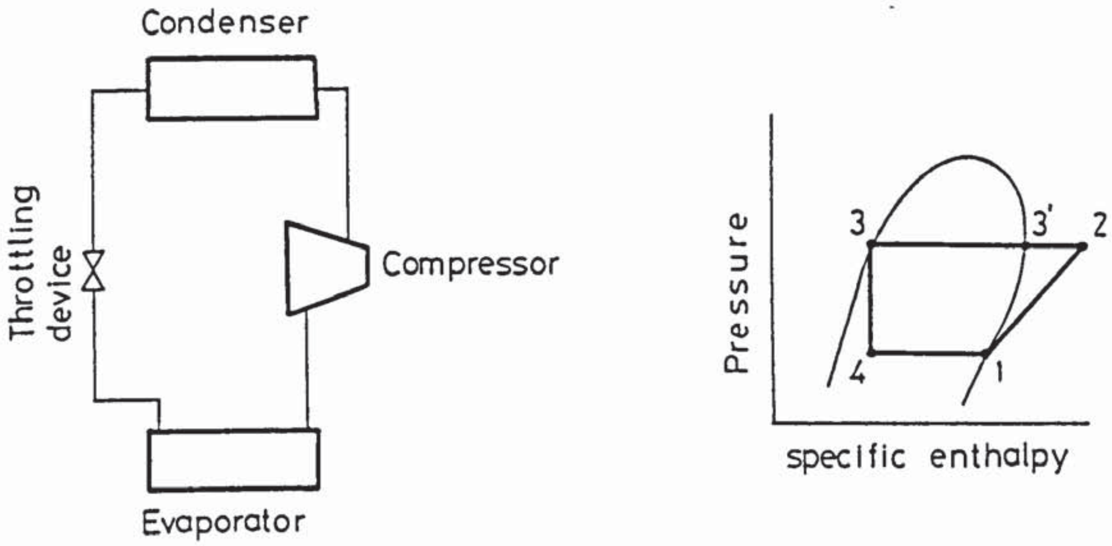


Figure 2.3 Vapour Compression Cycle

This cycle deviates from the ideal in that due to compression of the dry vapour (1-2) the working fluid is superheated, and must be cooled, at constant pressure, to state point 3^1, before isothermal condensation takes place.

2.3 SYSTEM CLASSIFICATION

To date there is no British Standard relating to heat pump equipment, and hence system classification is a matter of conjecture. However the method used in the U.S.A. [7] is proposed by Von Cube et. al. [8], where classification is according to the type of heat source, and the type of heat carrier. In addition to this a further distinction is suggested by Von Cube et. al. [8], by use of the terms heat pump and heat pump plant. The former relating to the refrigeration machine aspect, the latter being used when a secondary circuit is utilised between the heat source and the heat pump evaporator.

However, in general, it is only the heat source, and the heat sink, as per DIN 8900 [7], that are used to classify a system, for example, air-air; air-water. The first element referring to the heat source, the second to the heat sink. This is the classification used throughout this thesis.

2.4 HEAT SOURCES

All naturally occurring heat sources follow the external temperature fluctuations during the winter in a more or less attenuated way, such that the economics of heat pumps are diminished with falling external temperatures. Hence, for economic operation Von Cube et. al. [8] suggest the following:

a) A small temperature difference between the heat source and heat sink.

b) A heat source temperature as high as possible.

c) The cost for tapping the heat source must be minimised (less than 15% of the total cost of the plant).

d) The heat transport medium should not adversely affect the heat exchangers (corrosion-contamination-freezing).

e) The heat source freely available.

The most commonly used heat sources are:

i) Atmospheric air

ii) Ground water

iii) Surface water

iv) Soil

v) Solar heat

vi) Process waste heat

Throughout the project reported in this thesis atmospheric air was used as the heat source, a restriction imposed by the sponsoring organisation.

Atmospheric air is an ideal heat source, since it is readily available even in the remotest of locations, and utilisation is relatively inexpensive. Units can be factory assembled, with the minimum of site installation problems, and are therefore ideally suited to mass production techniques, which lead ultimately to further cost reductions.

However, two main problems become apparent when this heat source is considered further.

a) Due to its low thermal capacity, air is subject to rapid temperature fluctuations, and is at its lowest temperature when most heat is required.

b) When heat is extracted from the air, its temperature falls in the vicinity of the heat exchanger, resulting in moisture condensing out of the air. At low ambient temperatures this condensate freezes on the face of the heat exchanger reducing the rate of heat extraction.

Past development to overcome these problems has been directed towards supplementary (back up) heating below a "balance point" and automatic defrosting when heat exchanger blockage becomes intolerable.

2.5 HEAT SINKS

The most commonly used heat sinks are water and air, the choice of which depends upon the application, and the consumer's preference. For process heating, water is normally used, whilst for space heating air is the most economical, as the temperature of the space approximates to the air temperature. In hydronic heating systems, the water temperature must be considerably higher than the required space temperature for natural convection and radiation to occur, which results in a reduced C.O.P. McMullen et. al. [4] indicate that underfloor heating is ideal for heat pumps since the required condensing temperature is only about 30^{o}C, and developments in Germany using plastic pipe in special concrete screed have minimised the amount of thermal storage by the concrete.

2.6 HEAT PUMP COMPONENTS

Although the functions of the major components of a vapour compression machine are fixed, for each component there are numerous alternatives.

These alternatives are discussed below.

2.6.1 Prime Movers

The majority of existing heat pump installations use electric motors to drive the compressor, but fossil fuelled engines are now being used more often for commercial and industrial applications.

Each has advantages over the other, but it is suggested by McMullen et. al. [4] that neither is ideally suited to all heat pump applications.

a) Electric Motor Drives

These are quiet, require minimal maintenance, and are relatively inexpensive. However, they are generally fixed speed machines, which makes it difficult to vary the heat pump output, and during start up require a period of high current, which may create problems for other electrical equipment nearby.

Since the efficiency of electricity generation is only approximately 30% the majority of the energy supplied to the power stations is rejected to the atmosphere, where it is degraded to ambient temperatures.

b) Fossil Fuel Fired Engines

Engine driven heat pumps have the ability to utilise the exhaust and coolant heat normally rejected to the atmosphere during electricity production, such that the ratio of heat output to fossil fuel usage (usually termed the "primary energy ratio" - P.E.R.) is considerably higher. In addition to this, Bush [9] indicates, that because proportionally less of the final heat originates from the low temperature heat source in comparison with electric driven heat pumps, variations in the heat source temperature are not so critical, and higher process temperatures can be achieved without any reduction in the C.O.P. of the vapour compression cycle.

Several British manufacturers offer engines with power outputs above 100 kW, designed to industrial specifications, the price of which far exceeds comparable electric drives, and the outputs of which are in excess of the requirements for small industrial and commercial applications [10], hence there is considerable emphasis on the development of automotive engines for heat pump drives.

Heat pumps often operate for over 3000 hours p.a., whilst engines for automotive use, in general, run for less than 1000 hours p.a., with regular maintenance the rule rather than the exception [11]. Fearon [12] speculates that an engine drive could never be quiet enough nor reliable enough to be used in production heat pumps. However, the high level of sound attenuation possible with modern techniques negates the problem of noise, and with regular maintenance automotive engines are extremely reliable.

For automotive applications, a high power to weight ratio is necessary; thus these engines are highly stressed. However, it is suggested by Masters et. al. [13] that when used for stationary applications operating below the maximum conditions, these stresses are much reduced and engine life is greatly increased. With any internal combustion engine, the maximum efficiency and minimum wear occurs when the engine is hot and running, without the frequent stopping and starting experienced with automotive applications.

In stationary applications electronic governors are recommended by Pegley et. al. [14], which are capable of varying the engine speed and maintaining the engine loading between 50 and 90% which is the optimum range. Gaseous fuels, such as natural gas, ensure a large reduction in carbon fouling of the lubricating oil, and combustion chamber, thereby increasing the periods between services. Spark plug life is improved by the use of electronic ignition.

One penalty of using natural gas is that it occupies a greater specific volume than vaporised liquid fuel, resulting in a reduction of maximum power by approximately 15% [13]. The heated manifolds, used to assist vaporisation of liquid fuels, reduces the density of the gaseous fuel further and so should be avoided. Again, due to density considerations, Pegley et. al. [14] suggest that the air intake should be as cold as possible.

Electronic ignition is recommended by both Masters et. al. [13] and Pegley et. al. [14] which, under normal conditions, requires no maintenance, and besides improving fuel economy, enhances plug life and cold-starting. Fitting a bulk oil tank, which gives no weight penalty for stationary applications, and adding automatic water top up, further extends the service interval.

The major problem of natural gas as an engine fuel is the lack of lead content, which in gasoline acts as a lubricant. As the engine inlet and exhaust valves close on their seats there is a slight twisting action due to the valve spring. In the absence of lead lubrication this action causes the seat to be gradually worn away. This phenomenon is known as "valve seat regression", and can be countered by fitting Stellite valves and induction hardening the valve seats. The problem is more apparent in the exhaust ports and at high engine speeds, but it has been shown by Pegley et. al. [14] that at engine speeds up to 2500 rev/min there is little or no valve seat regression.

Current development is such that for automotive derived engines, service intervals are between 1000 and 1500 hours operation, and although extended life tests are still under investigation, figures between 12,000 hours and 15,000 hours are frequently quoted [13,15].

In Germany, engine driven heat pumps are at a fairly advanced stage, and current developments by Ruhrgas and the major vehicle manufacturers are tending towards a combined power and compression cycle from the engine unit, which, Rostek [16] suggests, will give increased engine stability, reduced noise and extended service intervals. It is understood, although no references have been found, that similar work is progressing at Cambridge University.

Current developments with internal combustion engines are towards limited cooled units [17,18,19,20] which offer up to 12% increase in thermal efficiency when turbocharged, and field tests are now under way in America using a derivative of the 185kW Cummins six cylinder in line diesel engine [18].

Also, it has been suggested by Burkhardt [21] that, because of ever increasing energy prices, gas engine driven heat pumps are now economically viable for domestic applications in Germany, using a 7kW engine.

Other fossil fuel fired engines are also under investigation. Dutram and Sarkes [22] appear to favour the Stirling cycle, whilst the Rankine cycle is considered the optimum by Strong [23].

At Cranfield Institute of Technology, research is towards the use of rotary sliding vane steam expanders (turbines) as the prime mover, driving a similar machine in a refrigerant vapour compression cycle. Walker et. al. [24] propose the use of coal fired Stirling engines combined with fluidised bed combusters.

c) Alternative Prime Movers

Reay et. al. [11] propose the use of a hydro-electric or even a direct water driven prime mover, while at Oxford University investigation into a vertical axis wind powered heat pump is under way.

2.6.2 Compressors

The requirements for a heat pump compressor are quoted by Von Cube et. al. [8] as:

a) Long service life of at least 25,000 hours.

b) Ability to tolerate high pressure ratios due to low evaporating and high condensing temperatures without overheating.

c) Should be insensitive to refrigerant aspiration.

d) Oil foaming must be eliminated.

e) Output control is necessary, but should not reduce efficiency.

However, in practical configurations numerous factors impede these requirements.

A proportion of the lubricating oil is carried out of the compressor with the refrigerant. Unless returned efficiently, the remaining oil becomes steadily diluted until premature failure occurs [12].

Most compressors require a dry suction vapour, as liquid slugs could result in serious damage. To ensure the elimination of these slugs the suction vapour is superheated. This not only elevates the discharge temperature but, due to the increase in specific volume, necessitates the use of a larger compressor than would otherwise be necessary.

The various compressors available for this type of system are:

a) Reciprocating compressors

b) Centrifugal compressors

c) Rotary vane compressors

d) Screw compressors

e) Scroll compressors

This project evaluates the rotary vane compressor used in the Denco-Ford "concept engine package".

Rotary vane compressors work with low compression ratios, but have high volumetric efficiencies. They are suited to high operational speeds, run smoother than reciprocating compressors and are compact, with low capital costs [11,25].

Read et. al. [26] suggest that these compressors are particularly suitable for use with automotive engines, since they can be directly driven, and run at a wide range of speeds with only minor variations in efficiency. A further advantage is the ability to run untroubled under slugging conditions.

The vanes may be located in the rotor or the stator and in either case operation is similar. The vanes, which are usually made from a reinforced phenolic resin, are free to move perpendicular to the axis of rotation, and are held in contact with the stator (or rotor) by the underblade pressure fed from the high pressure chamber. Sealing between the blades and stator (rotor) is effected by means of an oil film. This oil is injected into the compressor from a tank held at the discharge conditions.

When this project commenced there was insufficient knowledge of the operational characteristics of these compressors, hence extended durability tests are currently under way at British Gas Midlands Research Station [26].

2.6.3 Refrigerants

The choice of refrigerant is dependent upon a number of conflicting requirements, and so in general is a compromise.

It is shown by Reay et. al. [11] that for given evaporating and condensing conditions, C.O.P. is comparable for most refrigerant types, except when the condensing temperatures approach the critical point. However, variations in both vapour density and compression ratios are considerable, thus these are the main points to consider for working fluid selection. High compression ratios which result from low suction vapour pressures necessitate the use of large compressors and reduce volumetric efficiency [4,8,11]. High condensing pressures are detrimental due to the cost of components capable of withstanding extreme pressures. Thus the working fluid should have a reasonably high density, and only a small difference in evaporating and condensing pressures.

McMullen et. al. [4] suggest that viscosity and surface tension should be low to reduce system losses, but these factors inhibit the droplet formation necessary for condensation.

Chemical stability is important, because heat pumps operate at substantially higher temperatures than refrigerators. This can result in refrigerant breakdown at the compressor discharge port, and Reay et. al. [11] state that the products of this degradation are usually acidic, and so are harmful to mechanical parts.

Halogenated hydrocarbons are used extensively as refrigerants, and McMullen et. al. [4] suggest that this is because they are very safe, have a high chemical stability and do not readily decompose when in contact with metals at high temperatures.

Von Cube et. al. [8] consider these halogenated hydrocarbon refrigerants in depth, and Kruse [15] suggests that of these R12, R22 and R114 are the most popular refrigeration working fluids. When used for heat pump applications, they are working close to their limits of stability, and if these limits are exceeded, compressor failures can occur.

R502 is recommended by Von Cube et. al. [8] for heat pump use even though it has high operating pressures, since its low operating temperature enhances compressor life and oil stability. This is confirmed by Downing and Gray [27], who, when considering heat pumps installed at U.S. Air Force bases, show that maintenance costs and compressor failures, are considerably lower when R502 is used in preference to R22. At evaporating temperatures around $0^{o}C$ the capacities of these refrigerants are comparable, and so this improved reliaibility is achieved without any loss in performance. However, due to its high cost and poor availability in the United Kingdom, it was not evaluated during this project.

Currently the maximum condensing temperature for vapour compression heat pumping is limited to 100°C-120°C, due to the availability of suitable refrigerants. This is because the highest temperature in the cycle is immediately after compression and can be considerably higher than the condensing temperature.

The primary requirements for a high temperature refrigerant are:

a) The saturated vapour pressure must have suitable values in the required temperature range.
b) The highest pressure should be below 2 MN/m^2 (20 bar) to suit standard equipment.
c) It must be chemically and thermally stable.

Based on the above, Ekroth [28] suggests R114 is the most realistic choice for high temperature applications, a point confirmed by MacMicheal et. al. [29] in a gas engine driven, high temperature heat pump feasibility study. MacMichael et. al. [29] did however conclude that for centrifugal compressors R113 is a better working fluid, whilst Almin et. al. [30] suggest R133a is the optimum, since it is denser than R114 and lower superheat levels are possible.

Considerable work on alternative working fluids is now in progress, notably at Salford University, where members of the heat pump research group headed by Holland [31] have collated data for 21 potential working fluids.

The feasibility of condensing temperatures considerably higher than 120°C has been considered by Neil and Jensen [32] who propose the use of water/steam as the working fluid. Calculations based on isentropic compression from 120°C to condensing temperatures in the range of 150°C to 260°C indicate that water/steam has a much higher C.O.P. when compared with a number of organic fluids capable of these high operating temperatures. The only disadvantage is the high condensing pressure, of around 4.7 MN/m^2 (47 bar) at 260°C.

Kew [33] suggests that for a very high temperature heat pump, which he defines as condensing at temperatures in excess of 130°C, an absorption cycle, as outlined by Holland [34] is optimum. This is contrary to Moser [35] and Bauder [36] who suggest Fluorinol (Trifluoroethanol) holds out promise for vapour compression cycles condensing at up to 160°C, although its use is currently limited to expansion turbines.

Other important centres for high temperature heat pump development are Electricité de France [30] and Mechanical Technology Inc. U.S.A. [37].

2.6.4 Heat Exchangers

Heat exchangers are necessary to transfer heat from the heat source to the working fluid, and from the working fluid to the heat sink. The design of these heat exchangers depends upon numerous factors, and these are outlined below:

a) Reynolds number is probably the most important parameter according to Reay et. al. [11], since although high refrigerant flow rates in small diameter pipes give large heat transfer coefficients, they also increase to the pressure drop, hence any design must be a compromise.

b) The temperature gradient across a heat exchanger has a direct influence on the system C.O.P., and it is suggested by Von Cube et. al. [8] that the log mean temperature difference across a heat exchanger should be 3 to 4K.

c) Pressure losses must be minimised.

Heat exchangers can take one of two basic forms, depending on whether there is liquid-refrigerant heat exchange, or air-refrigerant heat exchange [4]. Specific designs are considered by Reay et. al. [11] and are listed below:

a) Shell and tube heat exchangers are used for liquid-refrigerant heat transfer, for either condensers or evaporators.

b) Finned tube heat exchangers are used for air to refrigerant heat transfer, and are of the cross flow construction, the flow of air being normal to the refrigerant flow.

c) Tube in tube heat exchangers. This is a true counterflow heat exchanger, with one fluid flowing in the inner tube, and the other flowing in the opposite direction in the annulus between the two tubes. Its main disadvantage is the large size required for practical applications.

d) Shell and coil heat exchanger. Is similar to the shell and tube arrangement, but is more compact, has good heat transfer coefficients due to the irregular shapes, but is not widely used and is consequently more expensive.

In addition to these, intercoolers are often used. These are refrigerant-refrigerant heat exchangers, and are used for heat exchange between the condensed liquid, prior to expansion, and the evaporated vapour, prior to compression, to ensure adequate subcooling, and superheating respectively.

2.6.4.1 Condensers

The two types most frequently used are:

a) Shell and tube condenser. Counter flow is suggested by McMullen et. al. [4] because of the necessity to desuperheat the refrigerant prior to condensation, and then to subcool the liquid before expansion. Because of variations in mass flow rate resulting from changes in operating conditions, a liquid receiver is required to give both storage space for excess liquid refrigerant, and system stability. If the refrigerant flows in the space between the shell and the tubes, the shell will act as a liquid receiver. Alternatively a separate liquid receiver would be required.

b) Finned tube condenser. Extensive work has been carried out on air cooled condensers by Carrington [38] and Blundell [39]. Both suggest that there is an optimum value for condenser face area, the number of rows, fin spacing, and air flow rates, and that the financial penalty is small compared with the return on investment. In addition to this Blundell [39] suggests that the maximum C.O.P. is reached for air off temperatures of 33^{o}C, and there is no advantage to be gained from circulating large quantities of air below this temperature. A separate liquid receiver is necessary for finned tube condensers.

2.6.4.2 Evaporators

For water source heat pumps the shell and tube heat exchanger is often used, and the refrigerant can flow either in the shell side or the tube side. The former is called a flooded evaporator. The oil separates from the refrigerant during evaporation, and sinks to the bottom of the shell. Oil drains are used to return the oil to the compressor. When the refrigerant flows in the tubes, it is termed a direct expansion evaporator. The evaporator must be sized such that the tubes are purged by the leaving vapour, which must be of sufficient velocity to carry the oil along as a mist, to the compressor.

Air source heat pumps use finned tube (direct expansion) evaporators. Carrington [38] and Blundell [39] suggest the performance of these may be optimised in the same manner as finned tube condensers, but the effects of condensation and possible freezing of the water vapour on the air side must be considered. McMullen et. al. [40], suggest that up to 15% of the heat collected by the evaporator is due to the latent heat of the condensing water vapour.

Blundell [39] suggests that evaporators designed using conventional refrigeration methods are acceptable, although the C.O.P. can be improved by reducing the air flow rate by up to 50%. Von Cube et. al. [8] confirm this latter point, and suggest the maximum face velocity should be in the order of 2-3 m/s.

Carrington [38] on the other hand suggests that both the air flow and heat exchange surface should be increased, and claims that this:

a) Increases the evaporating temperature.

b) Results in less frequent defrosting requirements.

c) Reduces the face velocity of the air, and hence noise.

The major disadvantage with finned evaporators is the formation of frost due to water vapour freezing on the surface, and the necessity to remove it. This problem is discussed in detail in Chapter 7. Richardson and Husker [41] have attempted to eliminate the frosting problem completely, and an ice clad evaporator is suggested. This consists of a smooth surfaced, solid state evaporator, utilising natural convection. However, due to the low heat transfer rates a large face area is required (approx. 0.7 m^2 face area per kW output), which precludes their use due to sheer physical size.

Alternatively, a buffer evaporator is suggested [42] which is contained in a hermetically sealed case of good thermal conductivity, with fins on the inner surface.

Inside this case there is a conventional finned evaporator, and a pair of fans. The fans force air over the evaporator and up to the top of the case. An air channeling system pushes the air down the side walls of the case where it absorbs heat from the outside air, completing the circuit. The air trapped in the case must be dry to prevent icing. Frost forms on the outside of the case but this is removed daily in extreme conditions by resistance heating elements.

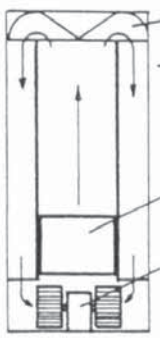


Figure 2.4 Buffer Evaporator [42]

2.6.5 Expansion Devices

The expansion device not only serves the important function of expanding the liquid refrigerant to low pressure prior to evaporation, but is also the traditional flow regulator in the refrigeration plant.

For small domestic refrigeration appliances capillary tubing is used as the expansion device, the diameter of which is determined by the most likely conditions encountered. It is unsuitable for heat pump applications as it is unable to cope with the wide range of operating conditions likely to be experienced. Hence thermostatic expansion valves (T.E.V.) are normally used:

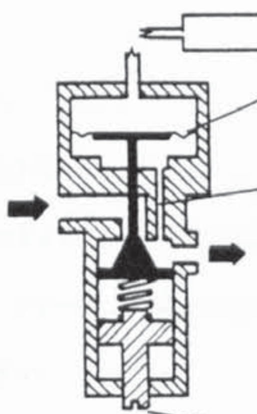


Figure 2.5 Thermostatic Expansion Valve [4]

The orifice in the expansion valve is restricted by a needle, attached to a diaphragm, one side of which is exposed to the refrigerant, the other to a vapour contained in a bulb. This bulb is attached to the compressor suction line, such that its vapour pressure is a function of the suction temperature. A spring is used to adjust the valve to its optimum setting. If the suction line superheat increases above its preset value, the bulb vapour pressure increases, depressing the diaphragm, thus allowing more refrigerant to flow, to regain the set condition.

Sizing of this valve is critical, since an undersized valve will result in underutilisation of the evaporator. The evaporating pressure will be reduced, resulting in lower evaporating temperatures and excessive compressor discharge superheat levels. Conversely, if the expansion valve is oversized, hunting will occur, such that the compressor will alternately receive highly superheated vapour followed by liquid slugs, due to the oscillatory action of the valve attempting to stabilise the conditions.

McMullen et. al. [4] consider alternative expansion devices and these are listed below:

a) Electrically actuated expansion valve.

b) Constant pressure expansion valve.

c) Float valve.

d) Electronic control valve.

2.6.6 Heat Pump Controls

Von Cube et. al. [8] suggest that water vapour may increase the enthalpy of air by over 50%, and is acceptable as long as the evaporator surface stays above $0^{o}C$. Below this temperature hoar frost develops forming progressively thicker layers until the evaporator becomes completely blocked and heat transfer ceases. This problem is particularly prevalent in the British Isles because, although winter temperatures are only moderately low, the relative humidity is constantly high.

Defrosting is currently accomplished by numerous methods. The most common ones are listed below, and are discussed in Chapter 7.

a) "Reverse cycle" operation.

b) Electric resistance heating.

c) Warm water or brine sprays.

d) Exhaust heat.

e) Hot gas bypass.

All the above methods require some form of controller to initiate and terminate the defrosting process, and it is suggested by Von Cube et. al. [8] that any defrosting controller should have the following features:

i) It must maximise efficiency by maintaining a relatively frost free surface.

ii) It must minimise the number of defrosting processes.

iii) It must be extremely reliable.

iv) Wind and sun should not affect the performance.

v) Fan failure and loss of charge should not initiate a defrosting process.

vi) Periods of defrosting should be as short as possible.

vii) A fail safe device is necessary.

viii) The controller must be easy to service.

The following are examples of control techniques available:

a) Time-temperature initiation. A defrosting process is initiated at fixed time intervals, during periods when frost formation is prevalent, and terminated when the refrigerant leaving the evaporator reaches a preset temperature, or after a fixed time period, whichever is the sooner. Switching off the fans during defrosting periods is suggested by Von Cube et. al. [8] to minimise the defrosting process time. Since it is difficult to estimate operating conditions McMullen et. al. [4] suggest that this method errs on the side of excessive defrosting processes.

b) Air pressure detection. As frost builds up on the face of an evaporator the air pressure difference across it increases. By measuring this pressure difference, defrosting can be initiated when a preset limit is reached. For an evaporator situated outside a building this method is sensitive to both wind gusts, and the possibility of leaves blocking the air channels. McMullen et. al. [4] suggest that a minimum air temperature sensor should be actuated before the defrosting process can be initiated, and that there is a finite time interval between subsequent defrosting processes.

c) Temperature differential. Reay et. al. [11] indicate that the temperature differential between the air and the refrigerant leaving the evaporator is more or less constant over a wide range of operating conditions, and is only increased significantly with frost formation. Thermistors or platinum resistance thermometers (P.R.T.), can be used to measure this temperature difference and so initiate a defrosting process.

d) Capacitance detection. It is shown by Buick et. al. [43] that for a small increase in thickness of ice formation between the plates of a capacitor there is a large increase in electrical capacitance which can be measured. However, experiments by Buick et. al. [43] indicated that although this system works, the long term effects of water penetration into the insulators causes the heating process to become shorter, until continuous defrosting occurs. Hence this system is as yet insufficiently reliable.

e) Radioactive methods. Since there are large amounts of hydrogen contained in ice and water, a neutron probe could be used to detect the presence of an ice film. However, McMullen et. al. [4] suggest that cost would be prohibitive, and that a radioactive source would be unacceptable in a domestic or commercial environment.

f) Microprocessor control. Mueller et. al. [44] suggest that the timed defrosting method is the best, and with a microprocessor it is possible to measure both temperature and humidity. Then based on laboratory tests the time cycle could be modulated to the optimum.

As a safeguard Mueller et. al. [44] suggest that the temperature difference between the ambient air and the refrigerant should be monitored to initiate and terminate the defrosting process in the event of a failure of the timing device.

2.6.7 Capacity Control

As ambient temperatures rise, the demand for heat falls, thus the ability to control capacity is essential, particularly as stop-start operation induces losses due to the effects of transient heating. Numerous options are available which are outlined by Tassou et. al. [45], but it is suggested by Masters et. al. [46] that a greater efficiency can be obtained if true speed modulation is adopted.

Heat pumps which draw heat from a natural heat source have additional capacity problems. During winter when most heat is required the temperature difference between source and sink is greatest, and heat availability is at its lowest.

Several authors [4,8,11,45] have advised that the heat pump should not be designed to provide the total heat load at the lowest anticipated operating condition, because at all conditions above this, it will be under-utilised. Instead, supplementary heating is recommended whereby the heat pump is designed to produce only a percentage of the heat load at the design condition, the remainder being supplied by conventional heating techniques.

Preferably this supplementary heating should be staged as shown in Figure 2.6, and the balance point should be somewhere between $-3^{o}C$ and $6^{o}C$ (depending on the literature consulted).

Goodall [47] considers the use of supplementary heating in some depth, and suggests that the actual energy used by

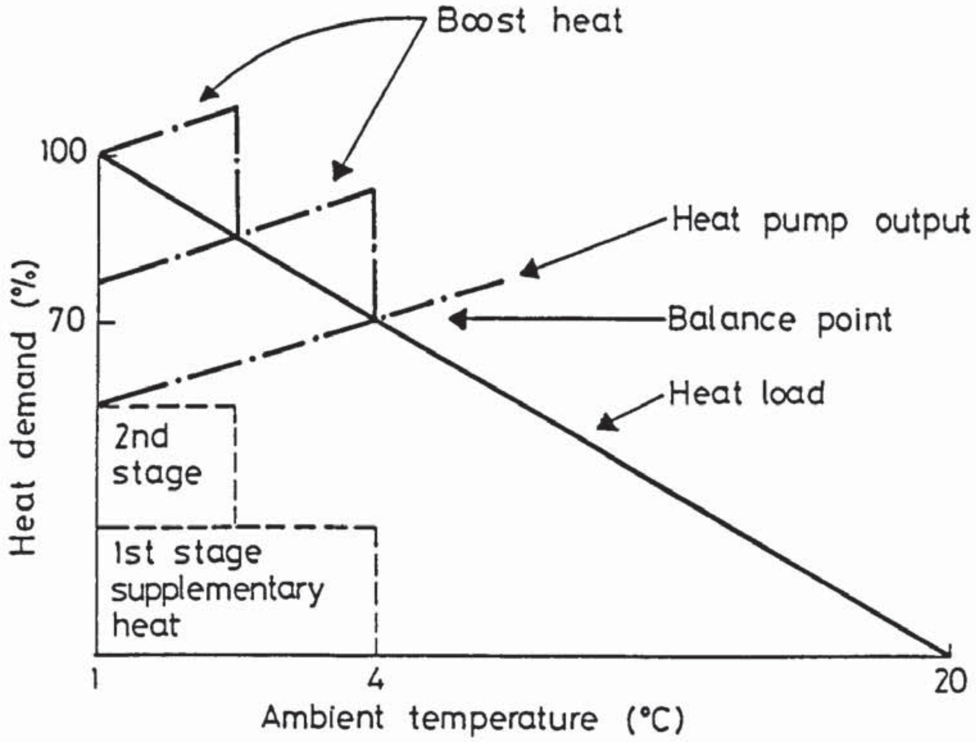


Figure 2.6 **Comparison Of Heat Demand And Ambient Temperature Using Supplementary Heating**

these heaters is considerably higher than predicted from degree day considerations. Possible reasons for this are that the back up heating will cut in on the indoor thermostat if the building suffers a sudden heat loss (due to doors opening) even though the heat pump may be capable of meeting the demand. To overcome this Goodall [47] suggests that supplementary heating is also linked to outdoor temperature, so that it will not operate above a preset ambient temperature.

It should be noted that Paul et. al. [48] state that supplementary heating is no longer used in Germany. Instead, the heat pump is switched off at low ambient temperatures and the boilers provide the full heat demand.

To control capacity by speed modulation, it is only necessary to monitor the indoor temperature. As the set point is approached the controller will regulate the speed by means of a proportional voltage such that the speed is reduced until the heat load is matched. When the controller has reduced the system to its minimum speed, stop-start operation is unavoidable, but in general this should only occur at the beginning and end of the heating season.

An alternative method of capacity control, is to use the dual unit method outlined by McMullen et. al. [4]. One engine driven heat pump, and one electric heat pump linked in parallel, both with speed controlled capacity modulation. In the spring and autumn the electric heat pump is used. As the ambient temperature falls the engine driven heat pump takes over, with its capacity for waste heat recovery, and in mid winter the two units work in unison. This system has the disadvantage of high capital costs.

Controls are also necessary to indicate overload conditions, and component failures, thereby preventing unnecessary damage. In particular Reay et. al. [11] indicate the following conditions must be avoided:

a) High discharge pressure (can cause structural damage).
b) High discharge temperature (corrosion of valves - refrigerant breakdown).
c) High suction pressure (damage to thrust bearings).

d) Low suction pressure (ingress of air - insufficient oil supply).

e) High compression ratio (mechanical damage of moving parts).

Simple interlocks can be used to sense these conditions and shut down the system.

The advent of microprocessor technology, and the comparatively low cost of micro-chips have led to the introduction of intelligent controllers for numerous processes. Mueller et. al. [44] suggest the microprocessor is an ideal device for controlling heat pump equipment and besides defrosting there are six functions it can perform, which are listed below:

a) Detect loss of refrigerant charge.

b) Detect loss of air flow.

c) Initiate a time delay prior to start up.

d) Detect compressor faults.

e) Monitor compressor crankcase temperature.

f) Provide fault diagnosis.

In addition to these six functions, Wilson et. al. [49,50,51,52,53] suggest that a microprocessor based device can also be used to optimise heat pump capacity by providing variable speed control for the condenser and evaporator fans, and the compressor. Although the use of variable speed electric motors is necessary, the increased cost is offset by the improved thermal efficiency of the plant. An improvement of over 10% in seasonal C.O.P. has been obtained by Tassou et. al. [49] compared with a conventional system with on/off cycling.

2.6.8 Lubrication

Oil is necessary to lubricate bearings, and to form a seal in heat pump compressors. However, compressors pump out some of this lubricating oil in a mist with the hot refrigerant vapour, so that oil circulates in the system. There are no difficulties with oil circulation from the condenser to the evaporator, since the oil is mixed with the liquid refrigerant, but high velocities must be maintained in the evaporator and compressor suction line to ensure adequate oil return with the vaporised refrigerant. Oil separators are often provided to minimise oil circulation, and these are fitted in the compressor discharge line.

Hughes et. al. [54] suggest that since oil is miscible with refrigerants (particularly R12) a proportion of the refrigerant will be contained as a liquid in the oil even after evaporation. If the oil becomes excessively diluted with refrigerant the lubrication properties will be impaired, and so high viscosity oils are recommended by Reay et. al. [11]. The effects on performance are somewhat blurred by conflicting reports. Fearon [55] outlines the work by Green [56] who suggests that heat transfer properties are improved if the oil content in R12 and R22 is between 0% and 3%. Above this level there is a decline, but it is not until in excess of 10% of oil (by mass), is contained in the refrigerant, that the heat transfer coefficient falls below that of a pure refrigerant. Hughes et. al. [54] suggest that particularly with R12 the evaporator capacity is reduced since the refrigerant mixed with the oil remains in a

liquid state and is unavailable to absorb the latent heat of vaporisation. In addition, it is suggested by McMullen et. al. [4] that the properties of this refrigerant-oil mixture are poorly understood, and the work of Bambach [57] is used in an attempt to clarify these properties.
In later work, Hughes et. al. [58] plot pressure-enthalpy diagrams of refrigerant-oil mixtures for oil concentrations of up to 15%, and suggest that it may be possible to overcome these problems by either reducing the oil-refrigerant ratio, optimising the superheat level, or by using oils and refrigerants which are less miscible.
The use of an oil separator is considered an unacceptable solution by Hughes et. al. [54], because up to 75% of the separated liquid could be refrigerant, and this would then be short circuited back to the compressor, thereby reducing performance and diluting the oil to such an extent that compressor faults could occur.
The results of this work by Hughes et. al. [54,58] suggest that R22 is a more acceptable refrigerant since it is less miscible with the lubricating oil than R12, and hence the above problems do not exist. This is contrary to the conventional refrigeration techniques reported by Von Cube et. al. [8], who regard miscibility as an advantage, as it improves oil circulation around the system.

Kew [33] suggests that conventional oils are unsuitable for high temperature heat pumps due to instability problems at the elevated temperatures, hence alternative oils are necessary, and current development is with synthetic lubricants for these high temperature applications [59,60].

The advantages of oil free operation are therefore considerable, and Hundy [61] has considered the problems of oil free operation with screw compressors. Although this goal has not yet been achieved, oil reduced operation (2%-5%) appears possible. Test results by Hundy [61] show that isentropic efficiency is comparable with conventional techniques, and the problems associated with cooling, lubrication and sealing are adequately (although not completely) solved by means of liquid refrigerant injection, into the compression chambers.

CHAPTER THREE

DESCRIPTION OF THE EXPERIMENTAL GAS ENGINE DRIVEN HEAT PUMP

The gas engine driven heat pump used for test purposes during this project was a pre-production model based on the Ford 1.6 litre automotive engine, and the Denco-Prestcold AGR 450* rotary compressor, employing a vapour compression cycle.

The heat load for the unit was provided by the Denco Air assembly plant, at their Plough Lane works, Hereford.

This chapter relates to the specific features of the gas engine driven heat pump which require detailed explanation. In addition, two production models of the same unit type were also studied (see Chapters 5 and 9), and minor differences between these and the pre-production model are highlighted in the relevant sections of this chapter.

3.1 GENERAL ARRANGEMENT

The assembly plant at the Plough Lane works can be divided into two main areas, (1) assembly and (2) paint spraying. The paint spraying area requires heat at a higher temperature than the assembly area, but the heat load is lower. Because of this the output from the gas engine heat pump is divided into two parts. The heat

* During the course of this project the patent rights for the AGR compressor were sold to Rotocold Limited, a member of the Syltone Group.

rejected by the condenser is utilised for heating the assembly area, by means of a ducted air system, whilst the waste heat recovered from the engine exhaust and coolant is used to heat the air in the paint spray booth (see Figure 3.1).

Because of the uneven temperature and flow distributions of air in ducts, and its associated measurement problems, the waste heat recovery and condenser heat are collected by a water system. For this reason the air ducted systems can be disregarded for the present exercise.

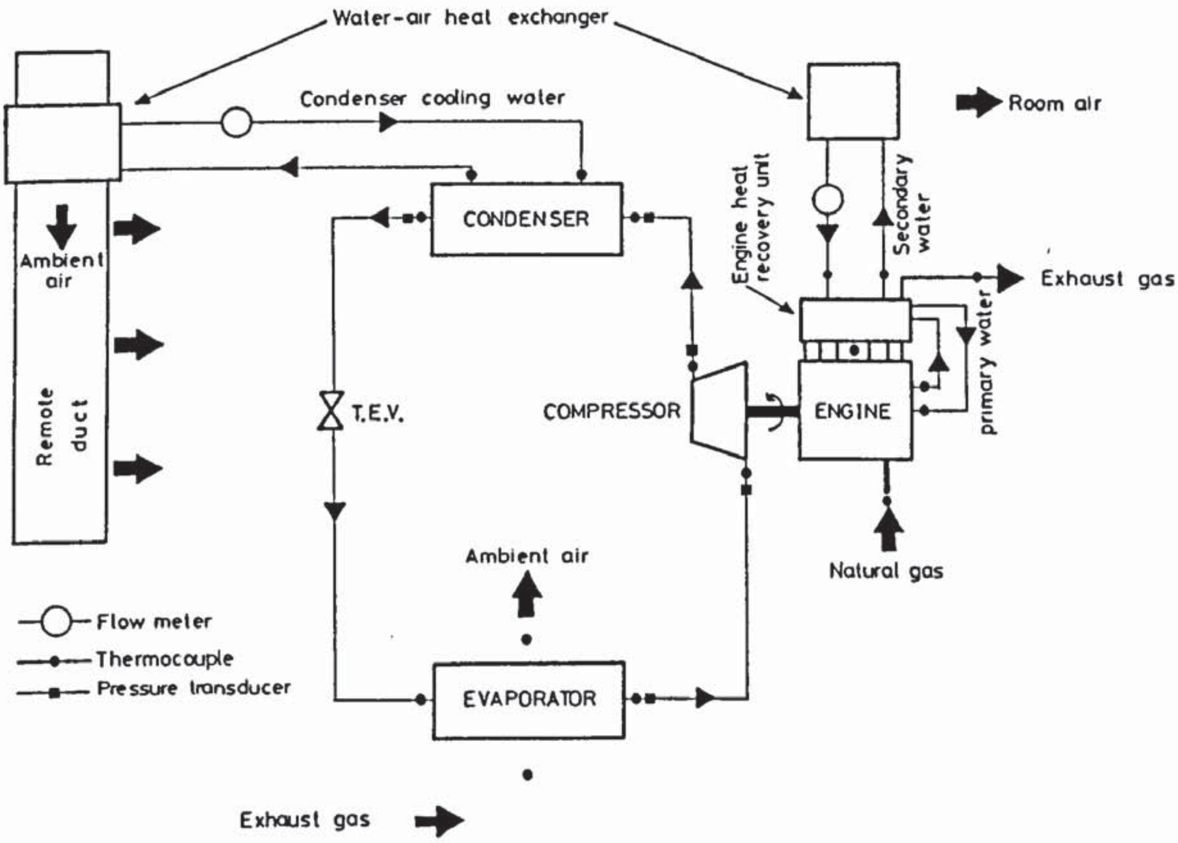


Figure 3.1 Schematic Diagram Of The Prototype Gas Engine Driven Heat Pump

3.2 NATURAL GAS PIPEWORK COMPONENTS

The statutory requirements for the fuel supply are outlined in the British Gas Publication 'Code of Practice for Natural Gas Fuelled Spark Ignition and Dual Fuel Engines' [62]. Figure 3.2 is a schematic diagram of the equipment fitted to the gas engine heat pump, and corresponds with these requirements.

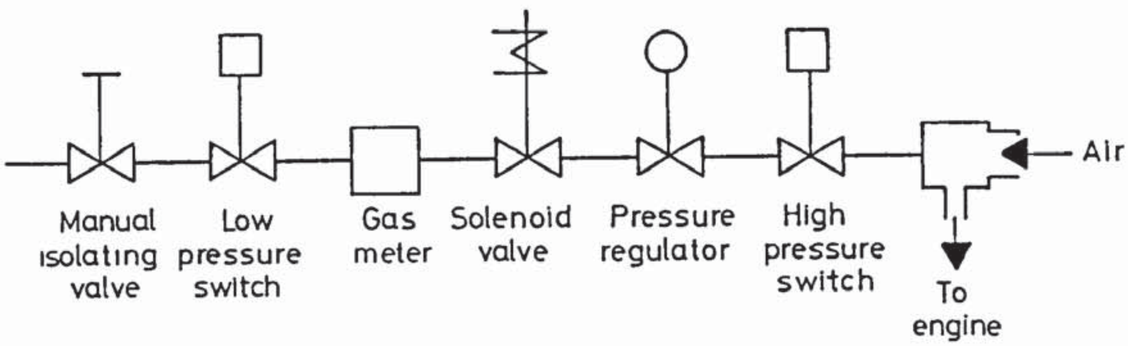


Figure 3.2 Natural Gas Train Fitted To The Prototype G.E.H.P.

The manual isolating valve is used to isolate the system during long periods of suspended operation, or when maintenance work is necessary. If for any reason the gas pressure should fall below that necessary for safe operation of the unit, the low gas pressure switch will cause the system to close down before damage occurs. The gas meter is not necessary under the statutory requirements, but is essential to measure the system performance. When the system for any reason is not running the solenoid valve is de-energised to prevent gas flow to the carburettor. Normal mains gas pressure is both excessive and prone to slight fluctuations which prohibit successful system operation, hence the regulating valve is fitted to control these conditions. If the normal mains gas pressure has to be reduced by more than 30%, surges may occur which cannot be compensated for by the gas pressure regulating valve. Thus a high gas pressure switch is fitted, which causes the system to close down in the event of a high gas pressure condition. The gas carburettor is necessary to regulate the flow of gas to the engine, and to provide the correct air/fuel ratio. It is an Impco type CA100-8, and is controlled by an electronic governor. The principle of operation of this governor is contained in the instruction and maintenance manual for the unit prepared by Hickman and Watkins [63].

3.3 PRIME MOVER

The prime mover is a standard Ford 2274E (Kent) 1.6 litre automotive engine, as used on the Escort model, converted to operate using natural gas. The main modification is an induction hardened cylinder head, and stellite valves, for the reasons outlined in Section 2.6.1.

A bulk oil tank located below the engine sump is fitted to all production units. This contains 35 litres of oil which is circulated to the engine sump by means of an oil lift pump.

3.3.1 Engine Governing

An Heinzmann electronic governor is used to regulate the speed of the plant, and to prevent an overspeed condition arising. Figure 3.3 is a schematic diagram of this governor.

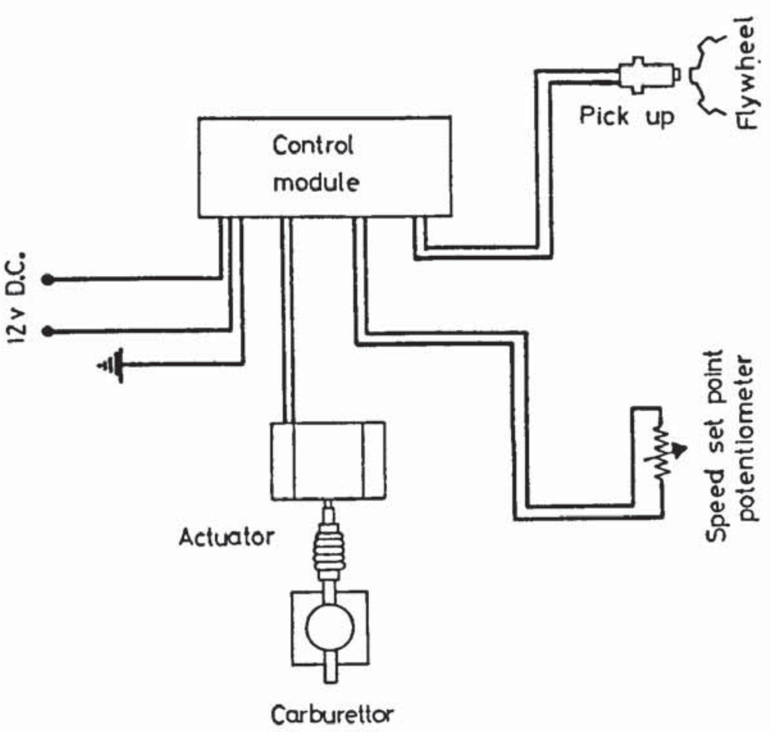


Figure 3.3 Schematic Diagram Of The Heinzmann Governor

3.4 ENGINE HEAT RECOVERY EQUIPMENT

Denco gas engine driven heat pumps are fitted with proprietary heat recovery equipment, which extracts heat from the exhaust gases, the engine coolant, and the lubricating oil.

3.4.1 The Serck Heat Recovery System

This comprises a shell and tube heat exchanger for coolant and exhaust heat recovery, and is designed to replace the conventional exhaust manifold. A supplementary oil cooler is located between the oil filter and its housing, and the oil is cooled by the primary engine water which is piped to this cooler.

Figure 3.4 is a schematic diagram of the equipment which is fitted to the pre-production experimental unit. However, initial performance testing indicated that this equipment was both inefficient and unreliable, so it was decided to use an alternative for the production models.

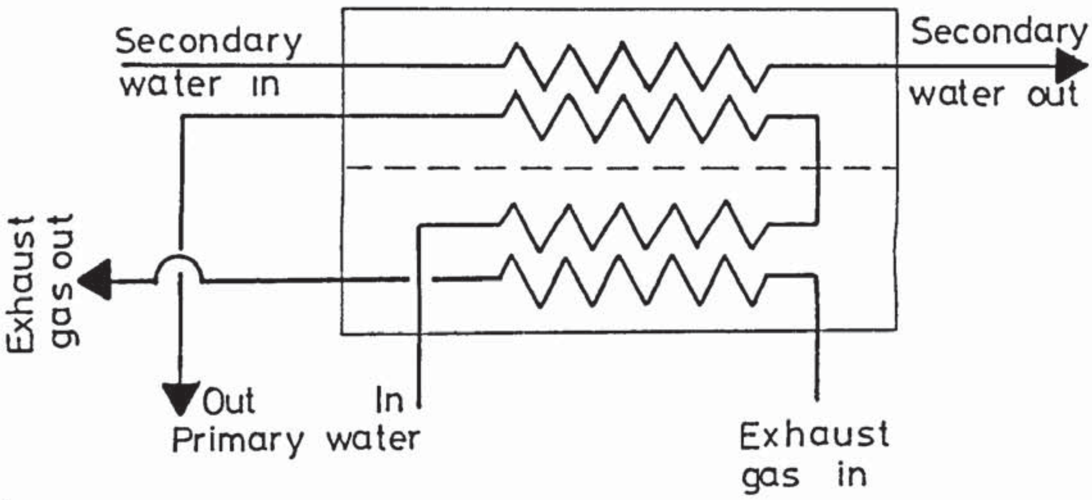


Figure 3.4 Serck Heat Recovery Equipment Schematic

3.4.2 The Bowman Heat Recovery System

A schematic diagram of this system is shown in Figure 3.5 and comprises:

a) Water cooled exhaust manifold

Exhaust gas temperatures in excess of 900°C [64] are experienced at high operating speeds, and so the manifold would glow red if it were not cooled. In automotive applications the airflow across the engine is sufficient cooling. However for stationary applications it is essential that the manifold is water cooled, and on the Bowman system the primary engine water is used for this purpose.

This water cooled manifold also incorporates a header tank to allow for expansion.

b) Shell and tube exhaust gas heat exchanger

The exhaust gas temperature may be lowered to below 60°C before condensation of the products of combustion occurs. The Bowman system utilises an exhaust gas to primary water shell and tube heat exchanger to approach this condition.

c) Shell and tube oil cooler

Cooling the engine oil not only provides additional heat output, but also maintains oil lubrication. Because of the high annual operating hours envisaged for heat pump applications a monograde oil (SAE 30) is recommended by the engine manufacturer to further improve the lubrication properties.

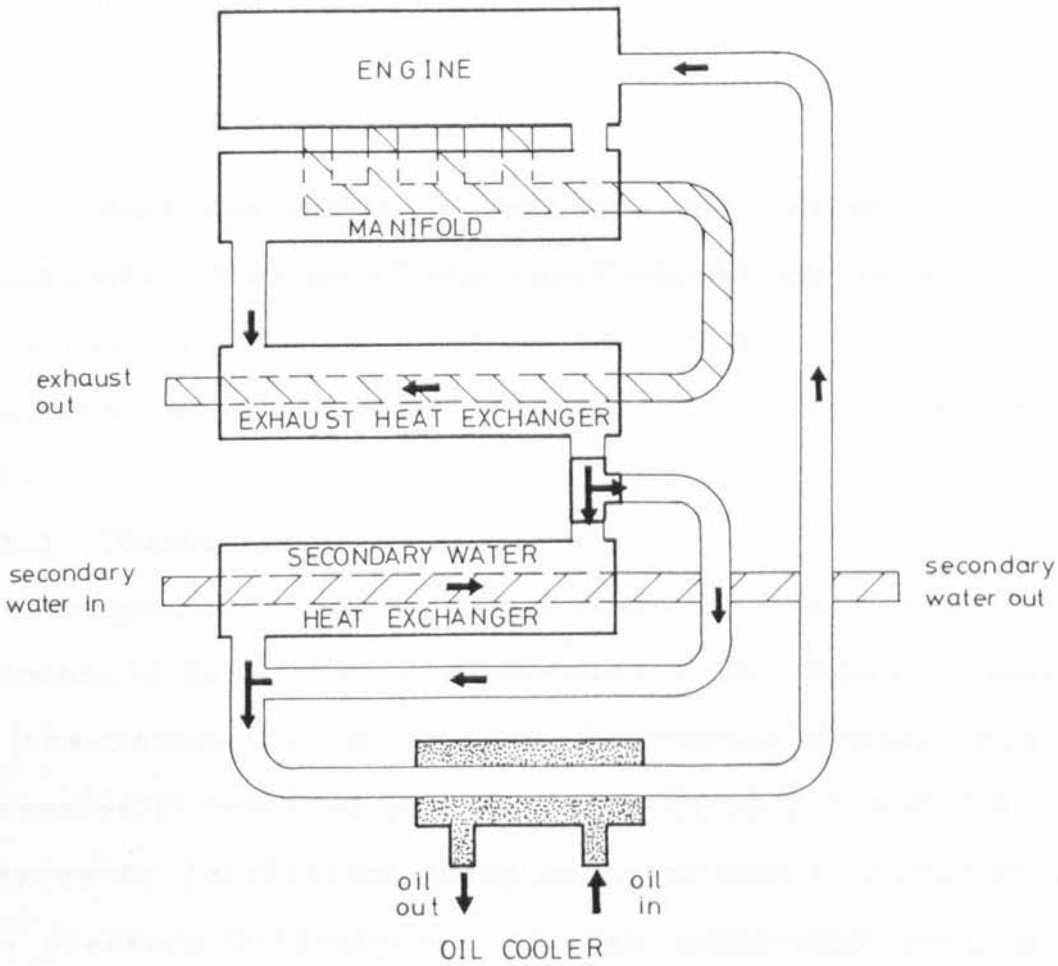


Figure 3.5 Bowman Heat Recovery Equipment Schematic

d) Shell and tube secondary water heat exchanger

The heat absorbed by the engine primary water from the engine jacket, the exhaust gases, and the engine oil, is transferred to the heat load by means of a secondary water heat exchanger.

e) Thermostatic by-pass

A thermostatic by-pass around the secondary water heat exchanger is provided in order to accelerate the engine pre-heat time, so that engine waste heat is not transferred to the heat load until the engine approaches its optimum operating temperature.

Both of these systems eliminate the need for the radiator and cooling fan normally associated with an engine of this type.

3.5 MAIN COMPONENTS OF THE HEAT PUMP SYSTEM

A schematic diagram of the experimental gas engine driven heat pump system is shown in Figure 3.6. This is based upon the vapour compression cycle utilising refrigerant R22.

3.5.1 Compressor

The compressor is of the rotary sliding vane type, originally developed by Denco-Prestcold. However, because of the necessity to measure compressor power, this was extensively modified as shown in Figures 3.7 and 3.8. Compressor lubrication is by means of Shell Clavus 68 oil. The pressure differential between compressor suction and the oil separator ensures an adequate supply of oil to the shaft seals and bearings, and to the compression chamber where it creates an oil film between the vanes and the

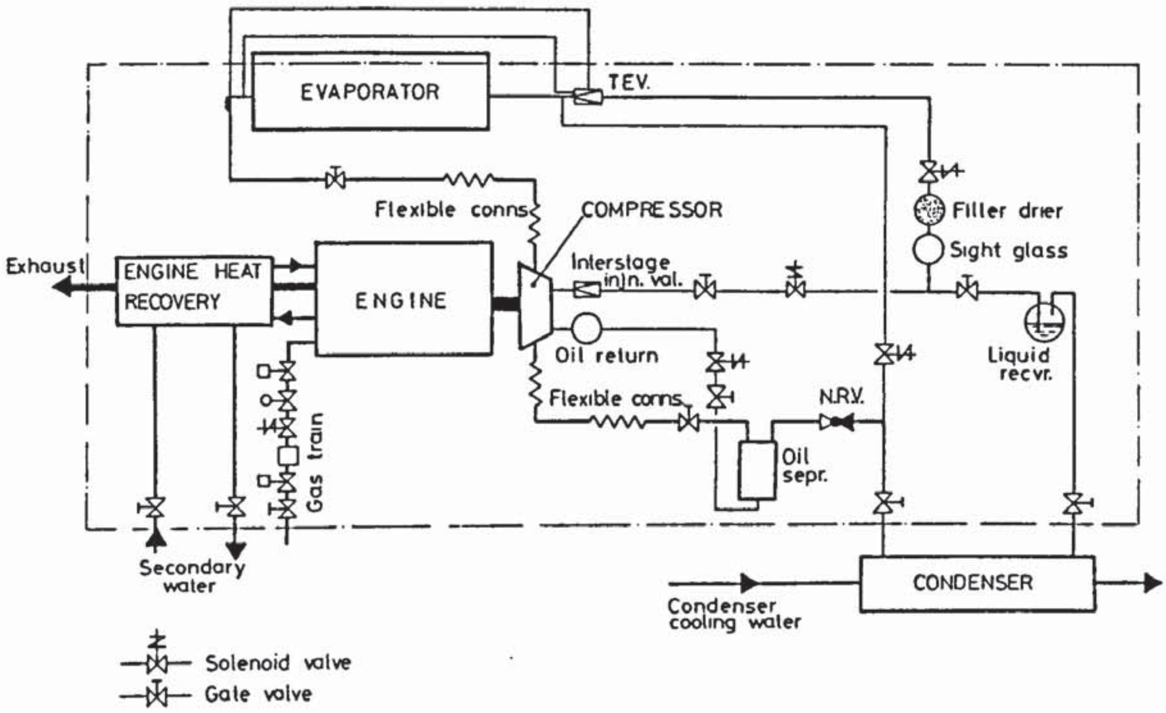


Figure 3.6 Schematic Prototype Gas Engine Driven Heat Pump Module

walls of the compression chamber, effectively sealing the individual cells.

Compressor cooling is essential to prevent damage to the seals and vanes. This is achieved by injecting liquid refrigerant from the liquid receiver into the compression chamber during the compression process. This liquid refrigerant is evaporated, absorbing its latent heat from the work of the compression and it effectively reduces the temperature of the discharged vapour, compared with operation without liquid injection. The quantity of liquid refrigerant metered to the compressor is controlled by means of a thermostatic expansion valve monitoring the compressor discharge temperature. The maximum discharge temperature recommended by the manufacturers is 80^{o}C.

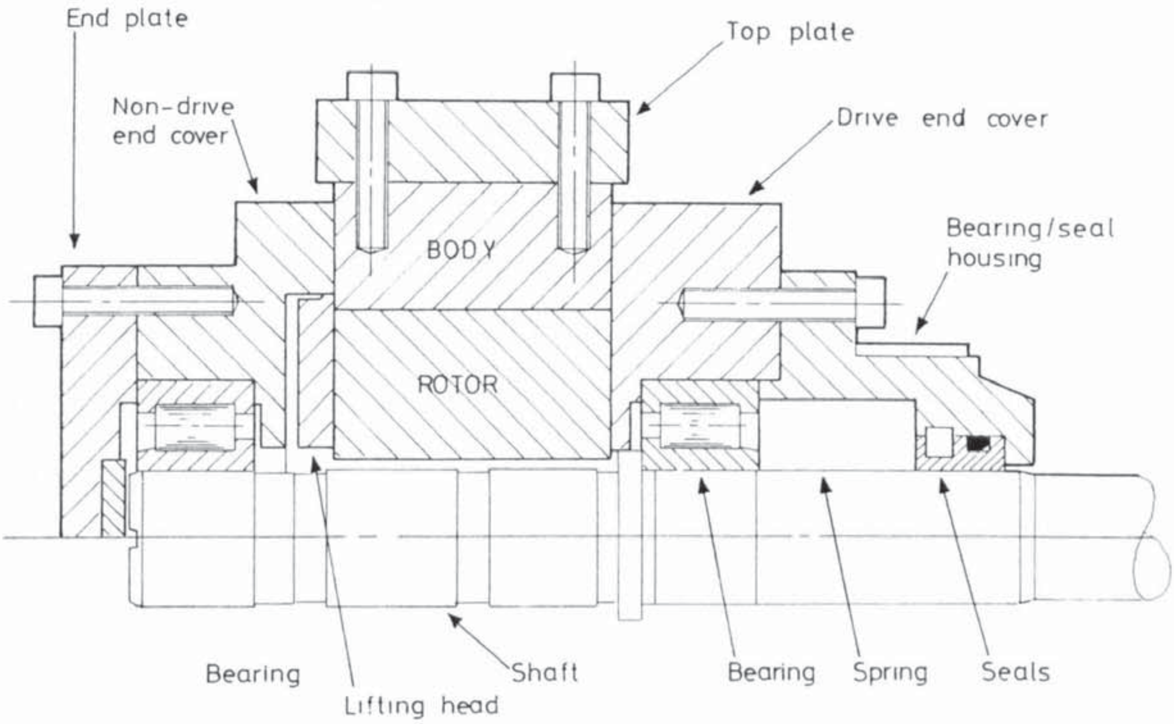


Figure 3.7 Section Through The Standard Rotary Sliding Vane Compressor

A solenoid valve is fitted in the liquid injection line before the liquid injection valve. This solenoid valve is normally open but is de-energised during plant shut down, to prevent liquid refrigerant from flooding into the compressor.

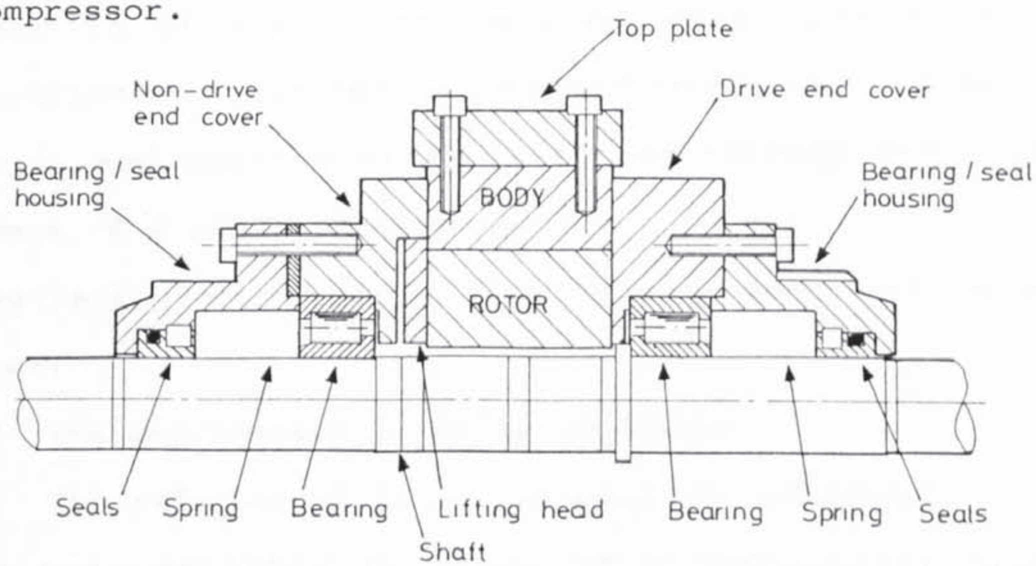


Figure 3.8 Section Through The Modified Rotary Sliding Vane Compressor

3.5.2 Oil Separator

Because of the large quantities of oil necessary for rotary sliding vane compressors (approximately 0.1 kg per kg of refrigerant) an oil separator is essential. Oil is carried from the compressor because of the high refrigerant vapour velocities, and unless adequate quantities are returned to the compressor, bearing failures and inter-cell sealing problems will result.

The oil separator used in this system is of the impingement type, and is located in the discharge line. Although very effective, oil separators are never 100% efficient and inevitably some oil will find its way into other parts of the system.

A solenoid valve is fitted in the oil return line and this prevents oil flow to the compressor during plant shut down and performs a similar role to that of the liquid refrigerant injection solenoid valve.

3.5.3 Condenser

The condenser used on the prototype gas engine driven heat pump is of the water cooled, shell and tube type. Vaporised refrigerant is supplied to the shell side of the unit, and cooling water is forced through the tubes by means of a centrifugal pump.

Regulation of the water flow rate is essential to ensure that:

a) The refrigerant is fully condensed.

b) The refrigerant is not excessively subcooled.

This is achieved by means of a refrigerant pressure controlled valve whose function is to maintain a constant condensing temperature.

Due to the wide speed range associated with the gas engine driven heat pump, and because the heat is extracted from the cooling water by ambient air, this valve cannot maintain a constant condensing condition. However, temperature fluctuations are maintained within a 10K range.

3.5.4 Evaporator

The evaporator is the finned tube direct expansion type, and was sized according to the procedure outlined by McQuay [65] for heat exchangers of this type. The physical dimensions of this evaporator are contained in Appendix 5.

Forced convection of the ambient air through the evaporator is accomplished by means of two axial fans mounted in front of the evaporator.

The evaporator is mounted at the rear of the gas engine driven heat pump module, such that ambient air is induced across the engine-compressor unit before passing through the evaporator. Any convected heat which would otherwise be lost is therefore recovered.

3.6 CONTROLS - SAFETY DEVICES

3.6.1 Control Module

The control panel is located within the Plough Lane assembly area, adjacent to the watercooled condenser, and contains the following items:

a) Mains isolator

b) Pump motor/fan motor, starters and circuit breakers

c) Control circuit breaker, and local-off-remote key switch

d) Hours run meter

e) Tachometer

f) Engine control module

g) Numerous pilot lights

The engine control module is an Alwyn 405 three attempt start module fitted with an engine temperature gauge, an oil pressure gauge, an ammeter, and a local-off-remote reset key switch. In addition, engine over heating, low oil pressure, and failure to start shut down devices are also contained within it.

The control sequence is outlined in the operating and maintenance manual for the unit [63].

3.6.2 Defrost Control

Hot gas by-pass is used for evaporator defrosting, which can be initiated either manually, or by means of a timer. Throughout the test period the timed defrosting process was overridden and defrosting was initiated manually as necessary.

On the initiation of a defrosting process, a solenoid valve opens to allow hot refrigerant vapour to pass from the oil separator outlet directly to the downstream side of the thermostatic expansion valve, and into the evaporator.

Condensation of the refrigerant vapour is inevitable with this type of defrosting system. However, since the rotary vane compressor is tolerant of liquid in the suction line, damage does not result.

3.6.3 High-Low Refrigerant Pressure Switch

Since both high compressor discharge pressure and low compressor suction pressure are detrimental to system reliability, a pressure switch is fitted which shuts down the plant when either of these conditions occur.

3.6.4 High Compressor Discharge Temperature Sensor

High compressor discharge temperature is detrimental for the reasons outlined in Section 3.5.1. A thermistor is fitted to the discharge line and this initiates shut down when temperatures in excesss of 80°C are experienced.

In addition to the above, high and low natural gas pressure switches are fitted which cause the plant to shut down in the event of large mains gas pressure fluctuations (see Section 3.2).

3.7 MEASUREMENT TECHNIQUES

There are six variables to be measured in the gas engine driven heat pump system:

a) Temperature

b) Humidity

c) Pressure

d) Fluid Flow

e) Engine Speed

f) Compressor Absorbed Power

Figure 3.1 shows the location of this test equipment.

3.7.1 Temperature Measurement

The most convenient method for measuring temperature is by means of thermocouples. Although the error of these ($\pm$ 0.5K) is much higher than the error of platinum resistance thermometers ($\pm$ 0.1K) the cost is considerably lower. In view of the large number of temperature measurements required, it was decided to use thermocouples, and at the points where improved accuracy was required, numerous thermocouples were linked giving a thermopile effect as outlined by Jones [66].

Nickel-Chromium, Nickel-Aluminium thermocouples were used throughout these experiments, and a typical circuit diagram is shown in Figure 3.9. It should be noted that in the case of the cooling water for the condenser and the engine heat recovery system, it is the temperature difference which is of importance, and in these cases a thermopile was mounted across the supply and return pipes. A single thermocouple was connected to the supply pipe, to measure the fluid temperature for the evaluation of fluid density and specific heat capacity.

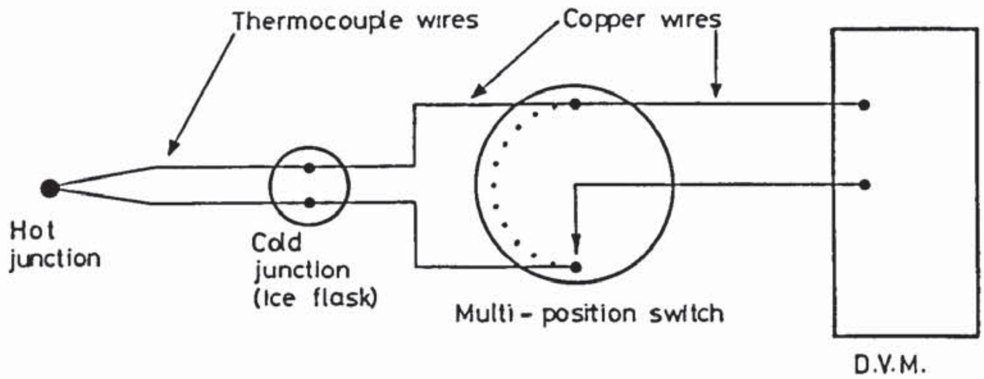


Figure 3.9 Thermocouple Circuit Diagram

Each thermocouple was mounted in the centre of the requisite tube by means of a copper pocket filled with oil to give a measure of the fluid rather than the tube wall temperature.

Three items were considered for the measurement of the thermocouple outputs:

a) An 'Edale'.

   This is a direct temperature measuring device which gives a direct reading of temperature and does not require the use of an ice point, the instrument itself being calibrated against ambient temperature. The output from this instrument is on an analog scale, and because of the large numbers of parameters to be measured, errors could result, due to fluctuations in the operating conditions based on a lengthy logging period.

b) Workshop potentiometer

Although very accurate, this instrument again has an analog scale and was therefore rejected.

c) Digital volt meter

The accuracy of this instrument is comparable with a workshop potentiometer, and in view of the speed with which readings could be recorded, this instrument was selected.

The conventional method to calibrate thermocouples is to immerse the couple together with a "standard" thermocouple into a zone which is at a uniform temperature, and to compare the readings [66].

The thermopiles were therefore calibrated by locating them in boiling and freezing water with calibrated thermometers.

Results of this calibration are contained in Appendix 3.

3.7.2 Humidity Measurement

The use of a wet and dry bulb sling thermometer for the measurement of humidity was considered unsuitable. An electronic humidistat was obtained, which had a proportional 0 to 100 mV output when supplied with 3.6 volts d.c. excitation. The accuracy of this instrument is quoted as $\pm$ 2% of reading, and was calibrated by the manufacturer.

The output was compatible with the digital volt meter and so was connected through the multi-position selector switch.

3.7.3 Pressure Measurement

In order to determine a refrigerant thermodynamic cycle it is essential for pressures around the system to be known. The use of Bourdon tube gauges was considered unsuitable for reasons of accuracy and ease of use. Hence absolute pressure transducers with a quoted repeatability of $\pm$ 0.5% F.S.D. were utilised. These gave a proportional 0 to 20 mV output when supplied with 10 volts d.c. excitation. The output from each transducer was linked through the multi-position selector switch to the digital volt meter.

The pressure transducers were calibrated by the manufacturer, but, each transducer had a zero offset condition. To determine this zero offset, the transducers were connected to the heat pump system, which was then charged with nitrogen to a pressure of 20 bar. The object of this was to strain the gauges prior to measuring the offset condition. The system was then allowed to stabilize at atmospheric pressure, before the offset conditions and the atmospheric pressure were recorded. The zero offset conditions are listed in Appendix 3.

3.7.4 Flow Measurement

a) Water flow measurement

Turbine flow meters were used to monitor the flow rate of cooling water to the refrigerant condenser and the engine heat recovery equipment. The frequency output from the turbine meters was converted to an analog signal within an electronic package supplied with the meter. An output terminal on this device gave a 4 to 20 mA output proportional to flow. This signal was passed through a resistor, and the voltage drop across the resistor was measured by the digital volt meter.

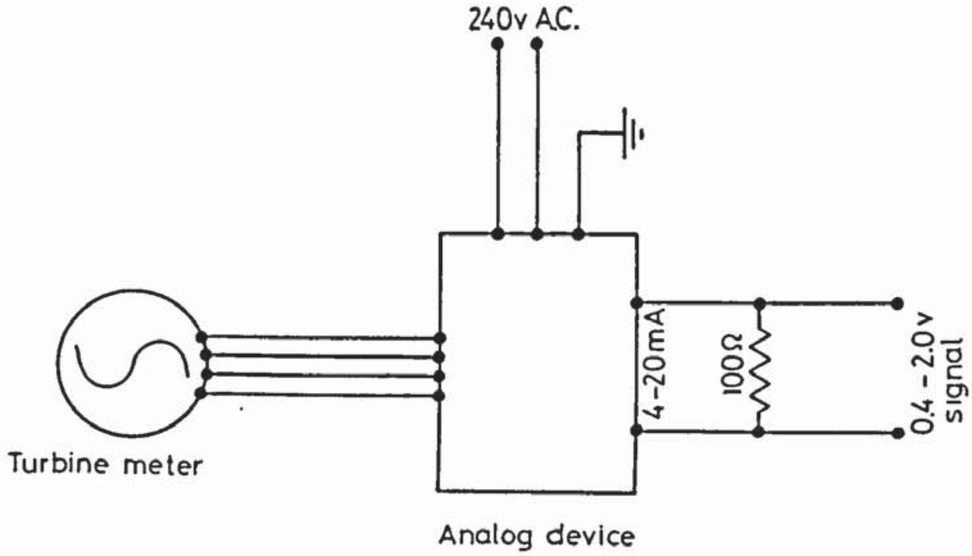


Figure 3.10 Wiring Diagram For The Turbine Flow Meter Measuring Cooling Water Flow Rate

b) Natural gas flow measurement

A calibrated totaliser type gas meter supplied by British Gas was used to measure the engine fuel consumption. The time taken for the unit to consume a fixed quantity of gas was recorded and the rate of fuel consumption calculated.

c) Refrigerant flow measurement

> The job was a job that just couldn't be done,
> But it had to be done, and he knew it.
> So, he tackled that job which couldn't be done,
> And found that he couldn't do it.
>
> Anon.

The necessity to monitor refrigerant flow becomes apparent if the efficiency of heat exchanger equipment is to be determined. Because of the phase change in the refrigerant cycle, and the inherent problems of two phase flow measurement, conventional equipment is unsuitable.

Various alternatives have been considered, and these are outlined below:-

i) Venturi tube/orifice plate/nozzle

Because of the pressure drop experienced with these types of instruments the tendency would be for boiling to occur, hence the vapour phase would vary between the pressure tappings and results would be very unreliable.

ii) Rotameter

Although suitable for liquid flow any vapour would give rise to unsatisfactory results, since the meter is calibrated for a given flow medium and any density or viscosity changes seriously affect the results.

In addition this type of meter has an accuracy no better than $\pm$ 3% full scale.

iii) Turbine meter

Again ideal for liquid flow, but vapour in the line would increase the flow velocity causing the turbine to 'race'. This could result in long term damage to the bearings and in the short term dubious results would be obtained.

iv) Ultrasonic meter

There are numerous types of these, but only the simplest type will be considered here. The principle of operation is that a sound-wave moves faster when travelling with the current than against it.

A soundwave is transmitted downstream to one receiver and upstream to a second receiver (both diametrically opposite the transmitter) and the time intervals recorded. Since the output is the unweighted mean velocity across the diameter, a calibration factor has to be applied, and the flow profile must be identical to the profile during calibration. This can never be guaranteed with only a single phase fluid, so results for refrigerant flow would not be reliable.

v) Vortex shedding meter

The principle here is that a 'bluff body' (awkwardly shaped object) placed across a flowing stream causes a succession of fluid vortices to be emitted from its trailing edges, which can be picked up by a variety of frequency sensing systems.

In an ideal fluid the ratio of the vortex frequency to distance between vortices is proportional to flow velocity.

This type of system results in high pressure losses, so refrigerant boiling may be a problem.

vi) Mass flow meter

Since the mass flow of refrigerant is independent of phase, this would appear to be the best alternative. However, most flowmeters measure flow velocity and liquid density and multiply to give mass flow.

A true mass flow meter would be required for this application. The National Engineering Laboratories (N.E.L.) in Glasgow were contacted regarding mass flow measurement. They recommended an instrument supplied by Lee Engineering, and made by Micromotion Inc. of U.S.A.

This instrument is based on Coriolis theorem of force vectors, and the instrument measures the phase shift from two sinusoidal wave outputs. N.E.L. [67] have obtained an error of $\pm$ 0.4% of reading using a fluid with up to 40% gas in liquid. This final instrument appears to be the optimum device to measure refrigerant mass flow rate, but the cost was more than half of the total instrumentation budget for the project and so could not be justified.

A turbine meter was evaluated, and although other workers [68,69] found this type of device to be satisfactory when located in the refrigerant liquid line, tests on the gas engine driven heat pump system proved to be inconclusive, when compared with the mass flow rate calculated from the compressor swept volume and suction conditions. Because of this it was decided to calculate the refrigerant mass flow rate from the compressor swept volume, suction conditions, and compressor volumetric efficiency as determined by Woolas [70]. (The method used is outlined in BS 3122 part 1 method D).

In the light of experience, it is suggested that the inconclusive results obtained, using the turbine meter in the refrigerant liquid line, could be due to changes in the mass flow rate at compressor discharge resulting from liquid refrigerant injection to the compressor, regulated by the thermostatic liquid injection valve monitoring the compressor discharge temperature (see Chapter 5).

3.7.5 Engine Speed

The engine speed was set at the commencement of each test by means of the variable resistance speed potentiometer fitted to the electronic governor and the tachometer. The speed was checked periodically using a stroboscope.

3.7.6 Compressor Absorbed Power

The power absorbed by the compressor was measured using strain gauges fitted to the shaft coupling the engine and the compressor. Leads from the strain gauges were passed through the centre of the compressor shaft and linked to a slip ring assembly mounted at the rear of the compressor. 10 volt d.c. excitation was supplied to the strain gauges and the output signal was measured using the digital volt meter. The strain gauges were mounted in a full bridge configuration, and were calibrated on an Avery torsional testing machine type 6609 CHG. The compressor was subjected to known loads for the full range of engine torque anticipated and at each load the strain gauge output was recorded. Results of this calibration are shown in Figure 3.11.

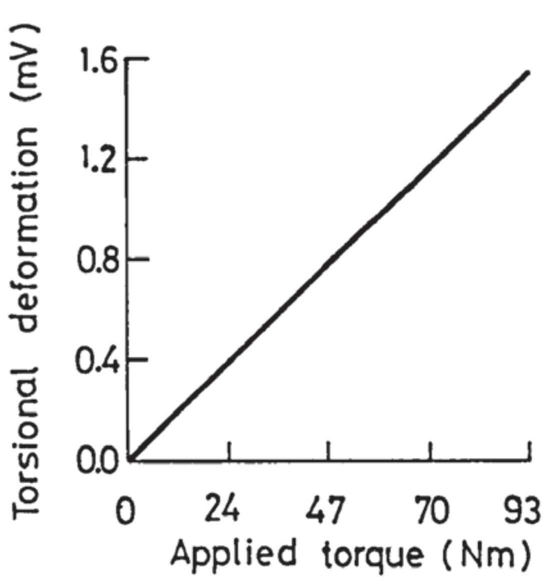


Figure 3.11 Characteristics Of The Strain Gauges Used To Measure Compressor Power

3.7.7 Data Logging

After approximately three quarters of the test programme had been completed a Solartron Orion data logger was obtained.

Transferring the measuring equipment to the data logger for the remainder of the test programme enabled the following improvements to be made:

a) Temperature measurement

Since the data logger had inbuilt cold junction compensation the use of the ice flask was no longer necessary.

b) Engine speed measurement

The data logger had the facility to measure frequency, so windings were placed around the high tension lead from the engine distributor, and these windings connected to the data logger.

With the circuit grounded a pulse was recorded each time the engine coil discharged. Only a simple scaler conversion was necessary to obtain the engine speed.

c) Natural gas flow measurement

The gas meter supplied by British Gas was converted to give a frequency output. A serrated disc was fitted to the rotating spindle on the meter and an optical switch consisting of a light emitting diode and a photo transistor receiver was used to generate a frequency signal.

Figure 3.12 shows the wiring diagram for this natural gas flow measurement device.

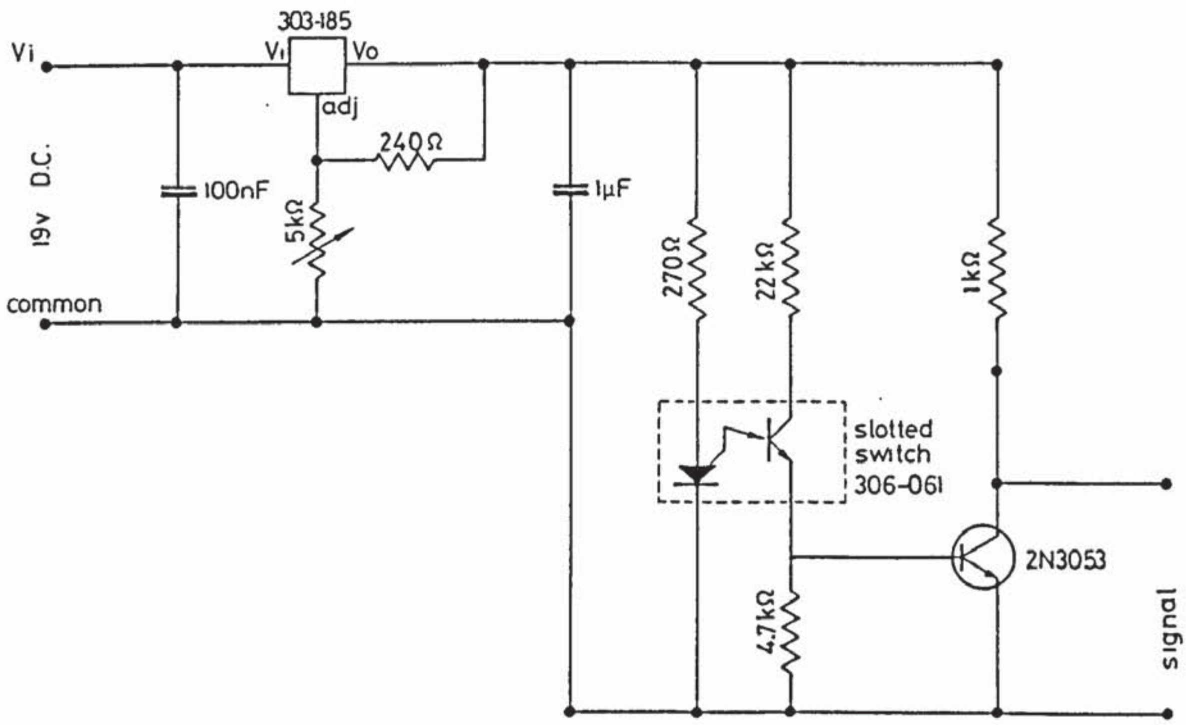


Figure 3.12 Wiring Diagram For The Gas Meter Conversion

CHAPTER FOUR

EVALUATION OF THE GAS ENGINE DRIVEN HEAT PUMP SYSTEM

4.1 STEADY STATE EXPERIMENTAL PROCEDURE

The object of these experiments was to determine the operating characteristics of a gas engine driven heat pump under steady state conditions; and to compare the performance with the engine and compressor manufacturers' published data. These steady state conditions were also required as a data base for subsequent transient experiments.

The plant was switched on and the engine speed regulated by means of a variable resistance potentiometer, to give a constant speed operating condition in the range 1500 to 3000 rev/min. This speed was checked using a stroboscope (Dawe Instruments type 1200E).

On reaching steady state, a defrosting process was initiated to clear any frost from the face of the evaporator. The gas meter reading was then noted. Approximately one minute was allowed for the compressor suction temperature to stabilize following this defrosting process, and then a full set of readings was taken.

Three sets of readings at each speed were obtained to give an average condition. These readings were taken at two minute intervals, which appeared to be the optimum. If an interval of greater than two minutes had been taken, frost formation on the evaporator would have been prevalent at low ambient conditions. This interval was also considered the minimum required to ensure that steady state conditions had been re-established. The gas meter reading and the time interval were recorded following each experiment.

The speed of the engine was adjusted, and checked using the stroboscope, and on reaching steady state conditions, the procedure outlined above was repeated.

Engine speed was subsequently modulated until results for the complete system speed range were obtained. To ensure that no lag in the system existed (particularly in the water circuitry) a complete set of results over the full speed range was obtained first by increasing the speed from 1500 to 3000 rev/min in 100 rev/min intervals and then repeated for engine speeds from 3000 to 1500 rev/min.

In addition to the above tests the air-flow across the evaporator was periodically checked with a proprietary anemometer (Airflow Developments turbine type), in order to detect any variations.

4.2 PRESENTATION AND DISCUSSION OF THE STEADY STATE RESULTS

The graphs contained in this section are based on empirical relationships derived from the experimental data contained in Appendix 3.

The performance of the refrigeration plant is considered first. This is followed by the engine performance, and finally the overall system performance is investigated.

Ambient enthalpy (kJ/kg of dry air) is used as a data base for the system performance, rather than ambient temperature, which neglects the latent heat content of water vapour in humid air. This latent heat is useful, since the water vapour condenses on the face of an air heated evaporator. The datum for ambient enthalpy was selected at $0^{o}C$ for dry air. The variation in ambient enthalpy was achieved by taking experimental data throughout a heating season, which covered the range shown on these graphs.

The evaporator air flow rate was kept constant throughout these experiments.

4.2.1 Refrigeration Plant Performance

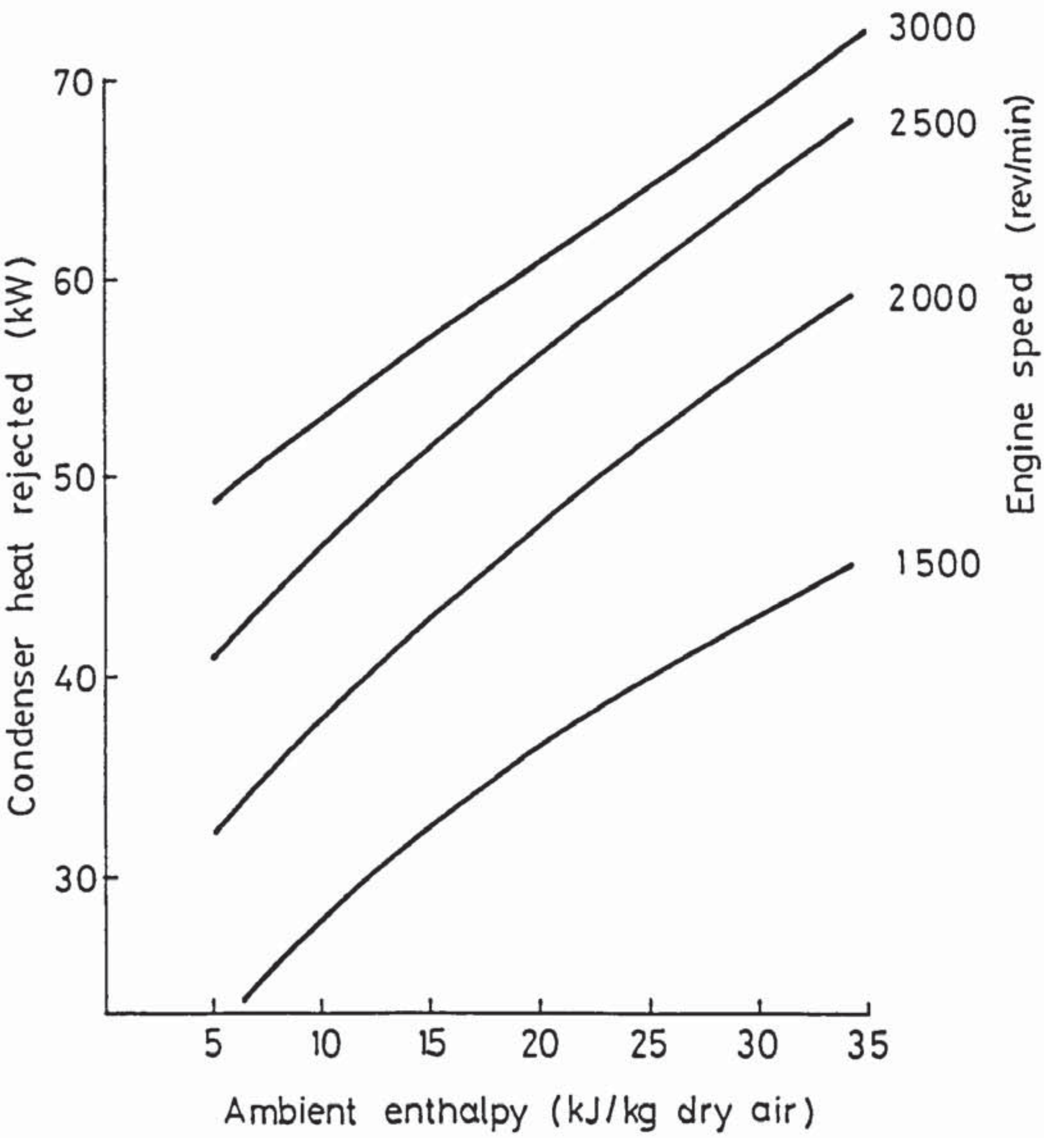


Figure 4.1 Condenser Heat Rejected v Ambient Enthalpy

Figure 4.1 is based upon the condenser cooling water characteristics. The trends of the curves are as anticipated. The condenser output increases with:

a) Ambient Conditions. The temperature lift between the heat source and the heat sink is reduced as the ambient temperature increases, and the heat available in the air is increased with increased ambient enthalpy.

b) Speed. As the compressor speed increases the refrigerant volumetric flow rate increases, which in turn increases the heat capacity rate of the refrigerant.

With both conditions a) and b) the refrigerant attempts to extract more heat from the air source, which has a fixed volumetric flow rate, causing the expansion valve to close in an effort to maintain equilibrium. This reduces the evaporating pressure, and hence temperature (Figure 4.7 page 70), which in turn reduces the refrigerant vapour density at compressor suction, and hence mass flow rate. This phenomenon results in a reduction in the rate of increased condenser heat rejection, with both increasing speed and ambient enthalpy, and is the limiting factor of the plant.

Figure 4.2 is based upon the refrigerant characteristics and thermodynamic considerations of the evaporator.

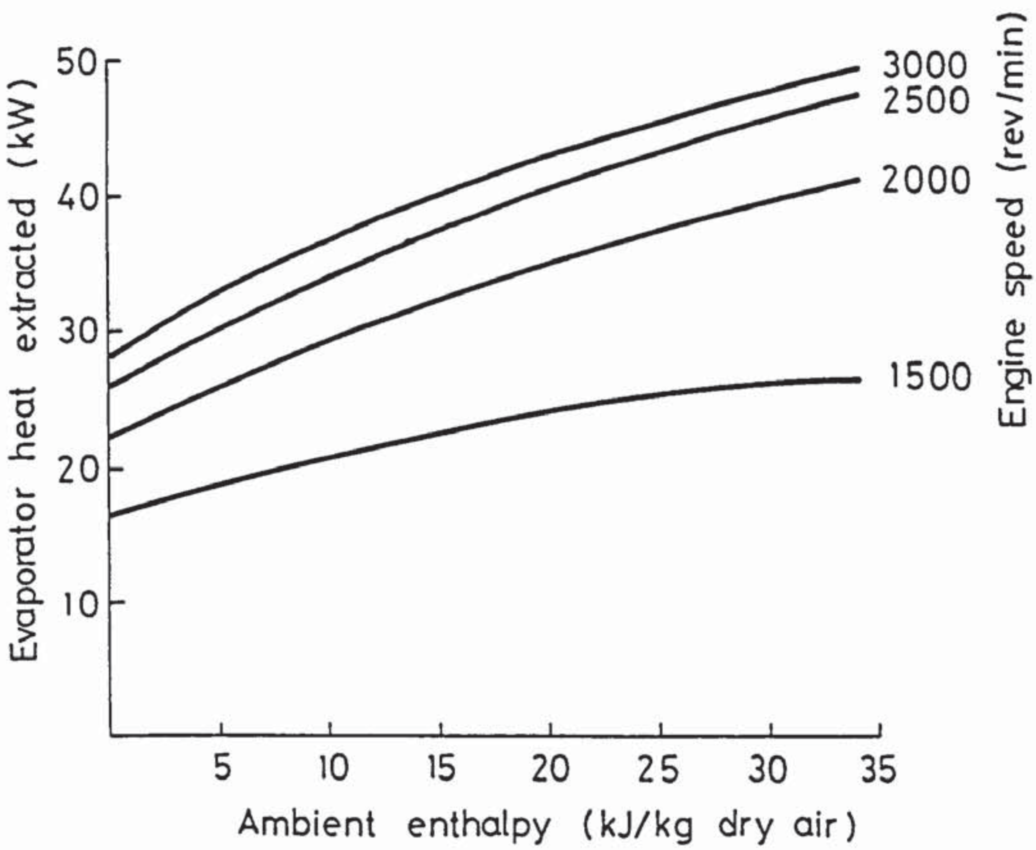


Figure 4.2 Evaporator duty v Ambient Enthalpy

As anticipated the shapes of the curves are similar to the curves presented for condenser performance, for the same reasons.

The increase in heat extracted at the evaporator is however less than the increase in heat rejected at the condenser. This is due to the additional mechanical work (in the form of compressor power) which is necessary to elevate the refrigerant to the condensing condition, as the compressor speed increases, and the temperature lift reduces.

Figure 4.3 shows the compressor power absorbed, which was measured using strain gauges mounted on the shaft coupling

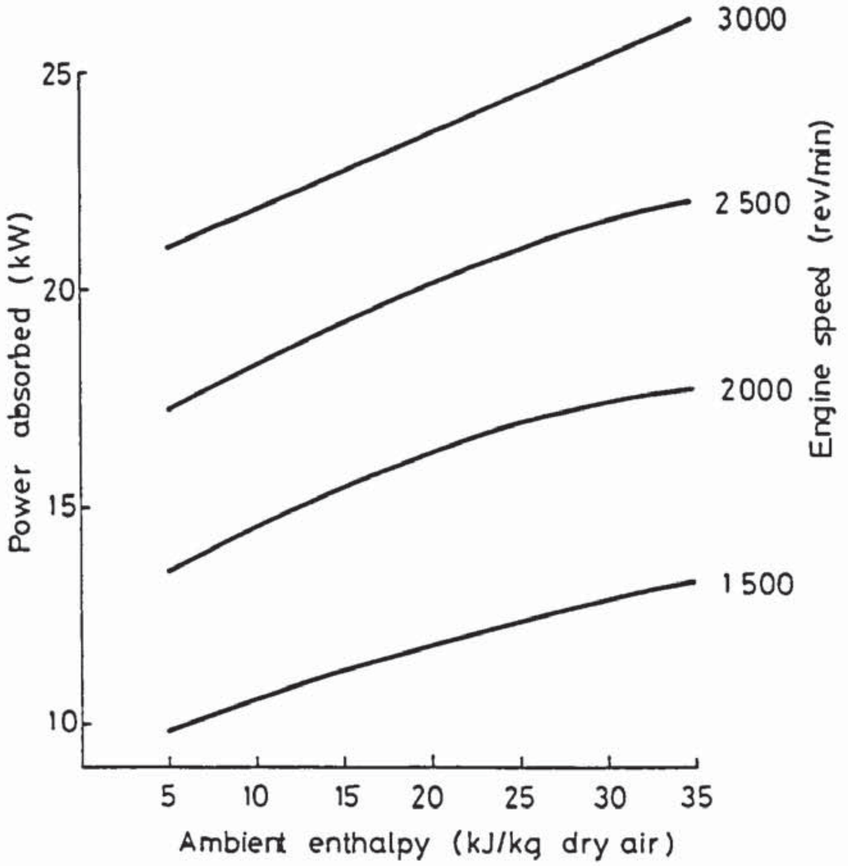


Figure 4.3 Shaft Power v Ambient Enthalpy

the engine to the compressor. The compressor shaft power increases with both increased engine speed and ambient enthalpy.

Figure 4.4 shows the heat rejected to atmosphere from the unlagged compressor discharge line. The rejected heat was calculated from measurements of the compressor discharge and condenser inlet conditions. None of the refrigerant pipework on the gas engine driven heat pump was lagged, as this was the normal policy of the sponsoring organisation. For air conditioning and refrigeration purposes this is not detrimental, since the condenser heat is rejected to waste. However, for heat pump applications it is the condenser duty which is important, thus any losses between

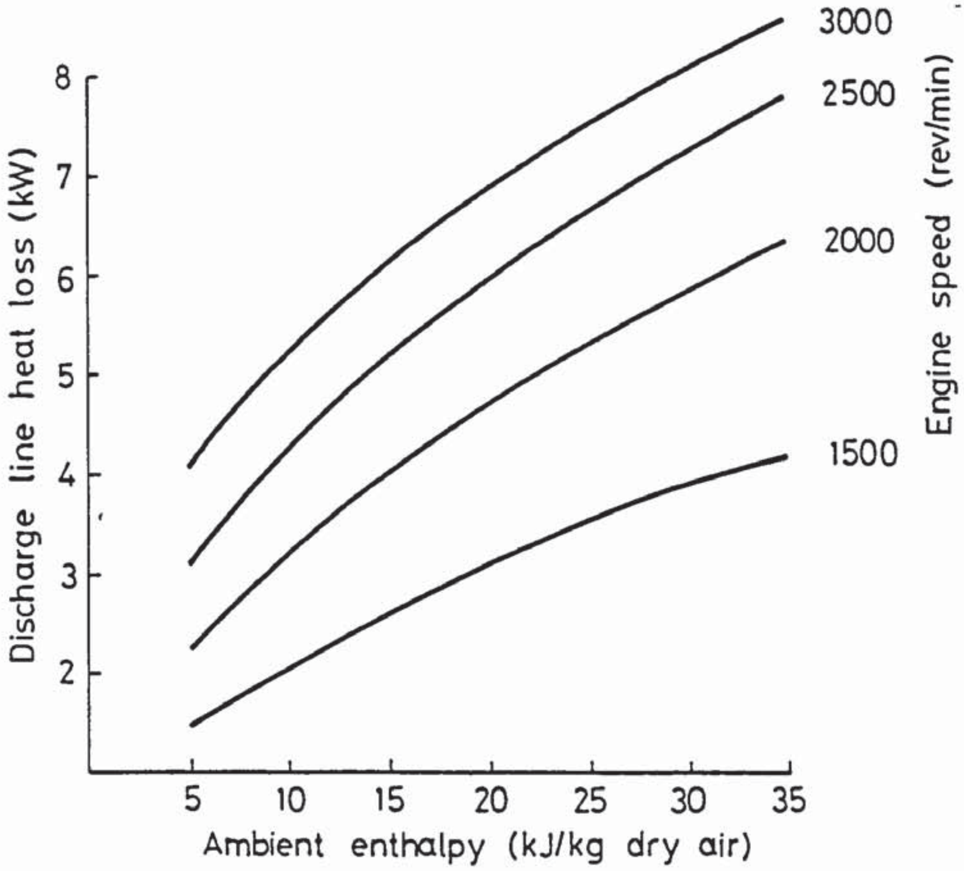


Figure 4.4 Discharge Line Heat Loss v Ambient Enthalpy

the compressor and the condenser result in reduced system performance. As the compressor discharge line to the condenser is over 15 metres long and includes the oil separator the sensible heat rejected to atmosphere before the refrigerant enters the condenser is considerable.

The coefficient of performance (C.O.P.) is shown in Figure 4.5 and is the ratio of heat rejected at the condenser to mechanical work provided by the compressor. This has been determined by combining empirical relationships from Figures 4.1 and 4.3 (Appendix 3.)

At high ambient conditions the C.O.P. is shown to be greater at low compressor speeds. At low compressor speeds the evaporating temperature is higher than at high

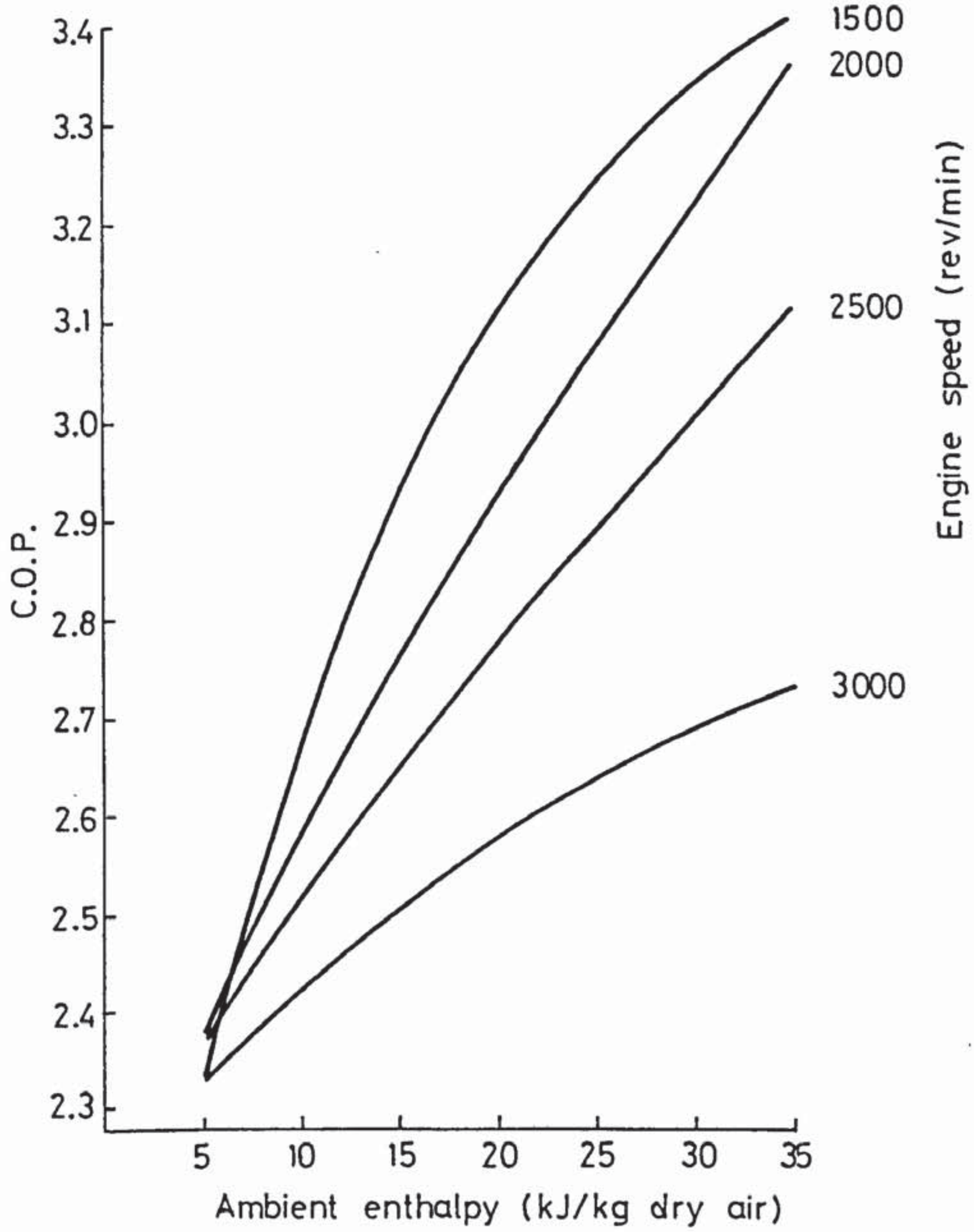


Figure 4.5 Coefficient of Performance v Ambient Enthalpy

speeds, because the refrigerant volumetric flow rate is lower. Less heat is extracted from the ambient air, and localised cooling of the air is reduced.

This higher evaporating temperature at low speeds effectively reduces the temperature lift between the evaporator and condenser compared with the high speed situation, reducing the necessary compressor power, and resulting in an increased C.O.P.

At an ambient enthalpy in the region of 5kJ/kg dry air, the variation in C.O.P. is small over a wide speed range. The reason for this is not apparent. However, tests show that at low speeds compressor vane chatter is experienced, indicating that the vanes are not fully extended. This allows some refrigerant flow back into the trailing chamber, which could be a contributing factor.

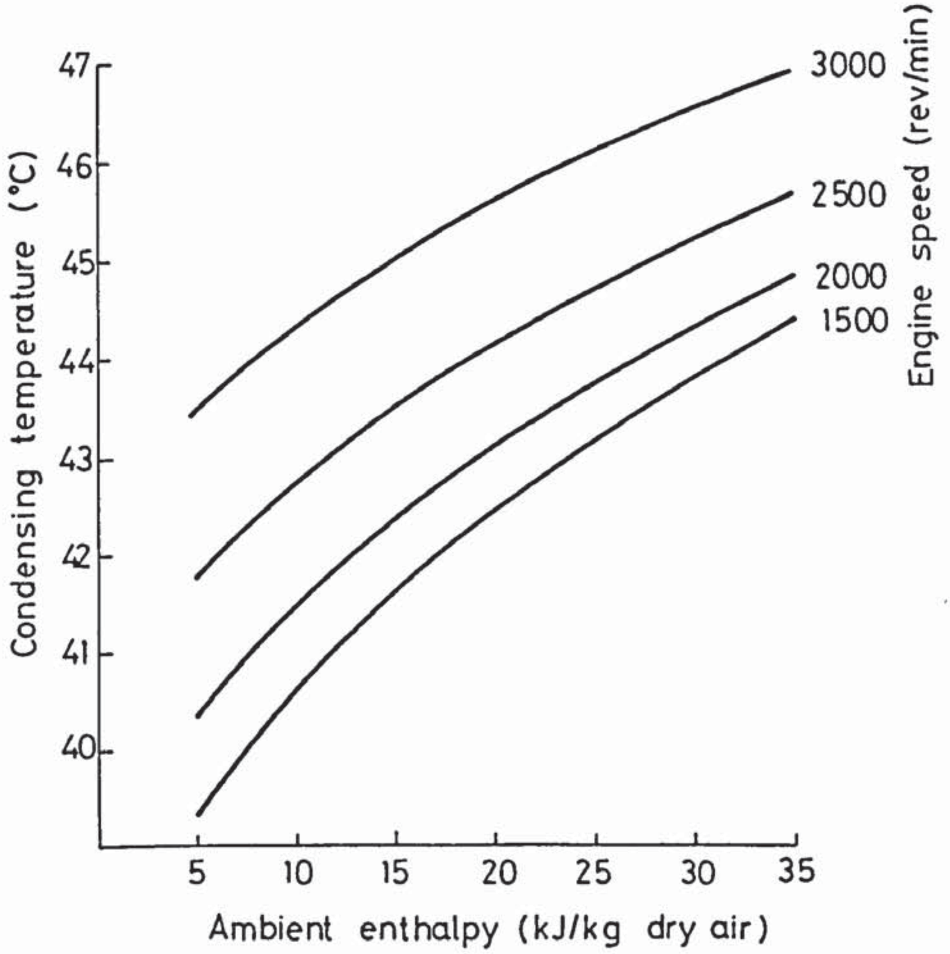


Figure 4.6 Condensing Temperature v Ambient Enthalpy

The condensing temperature is controlled by means of a condensing pressure regulating valve. The maximum condensing temperature experienced is 47°C, the minimum being just below 40°C. Thus, the fluctuation in condensing temperature has been maintained at ± 4K for a wide range of operating conditions.

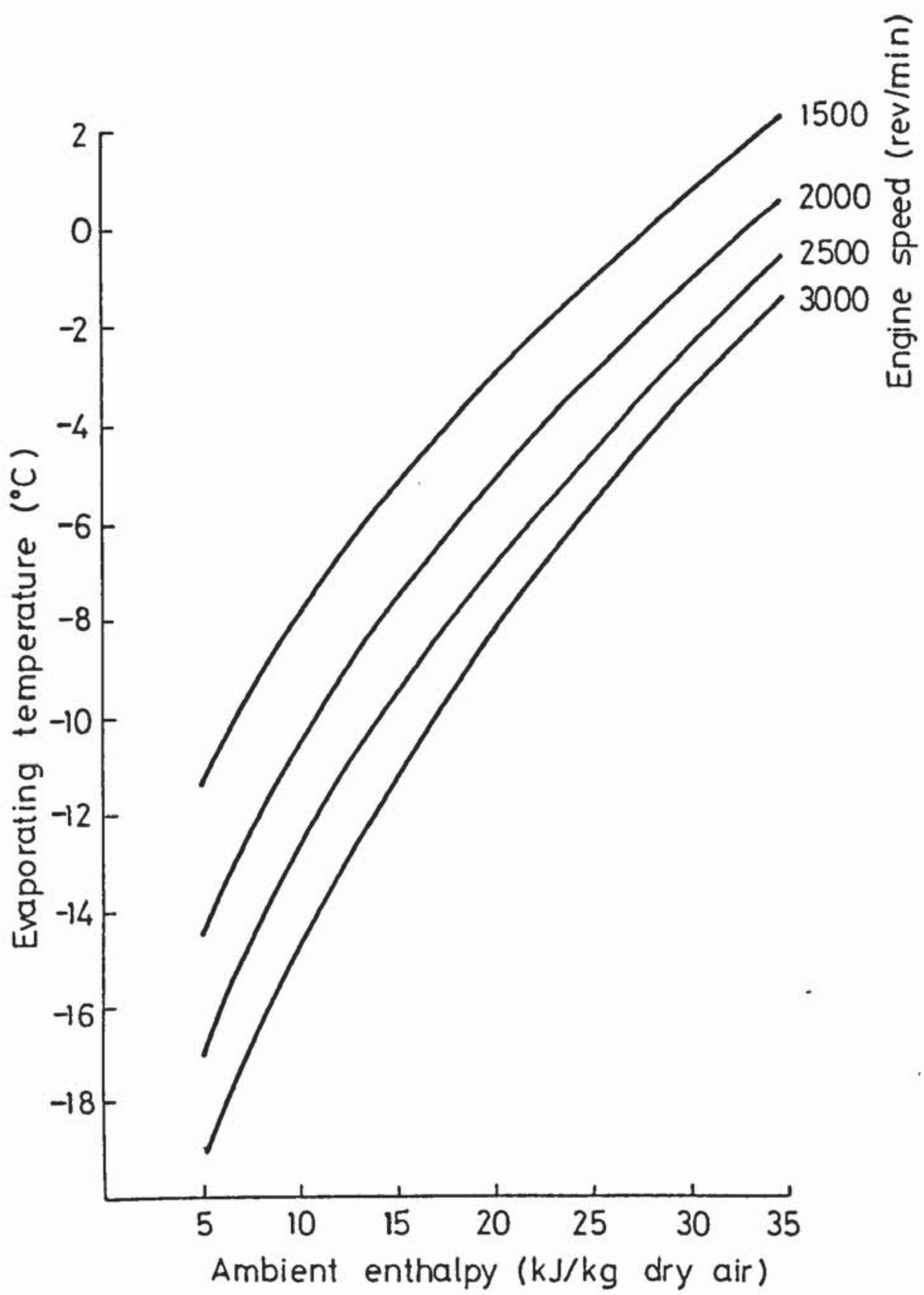


Figure 4.7 Evaporating Temperature v Ambient Enthalpy

The trends indicated in Figure 4.7 are as anticipated. The evaporating temperature increases with:

a) Ambient Conditions. As the ambient temperature rises, the evaporating temperature also rises, for the reasons previously discussed.

b) Speed. Because of the reduced refrigerant volumetric flow rate at lower speeds, the evaporating temperature rises as the engine speed falls.

From Figure 4.7 it appears that the evaporating temperature experienced on the prototype unit is very low, particularly at the high speed condition. A temperature approach (ambient air temperature - evaporating temperature) of 10K or less is necessary, according to Carrington [38].

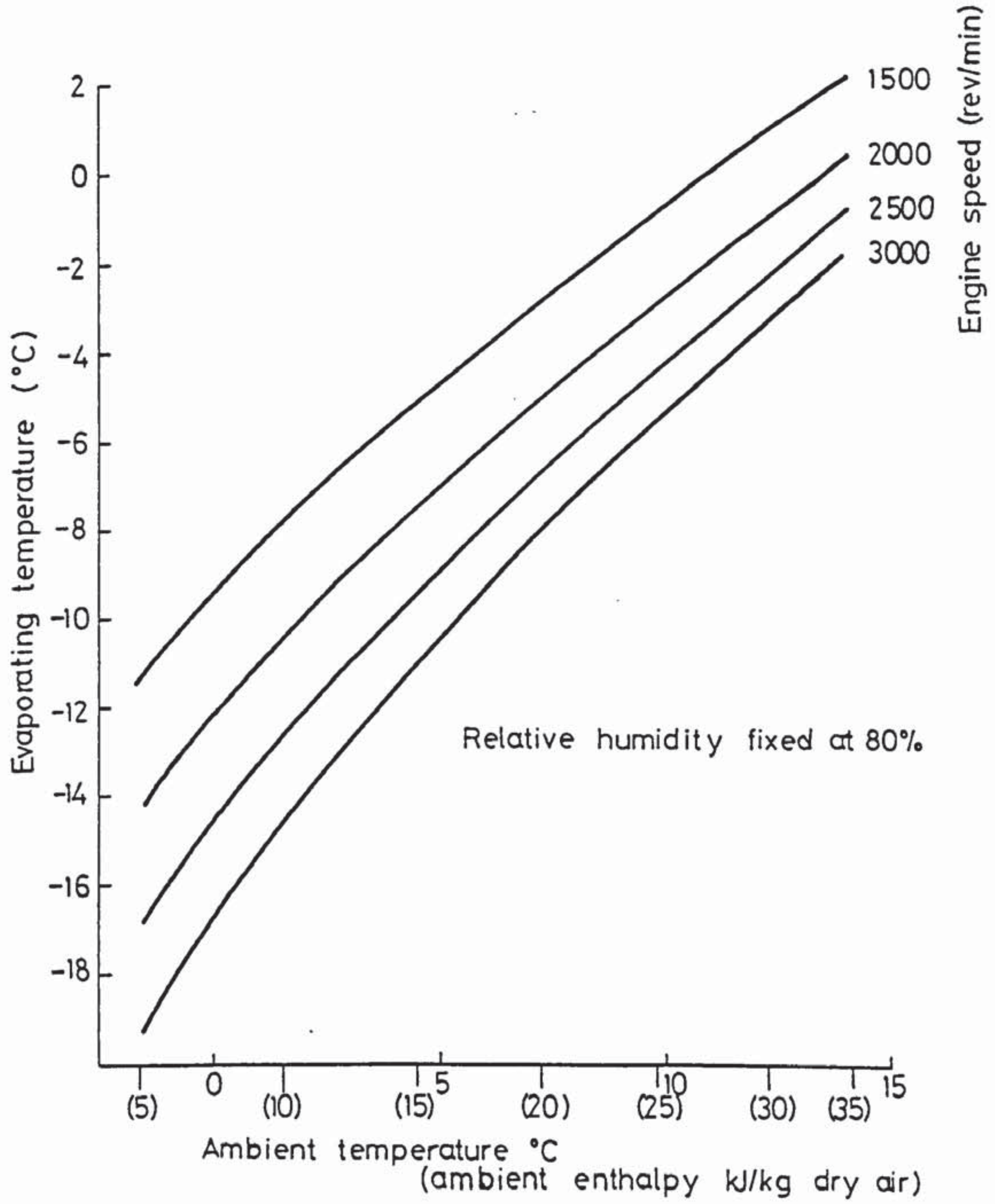


Figure 4.8 Evaporating Temperature v Ambient Temperature

In Great Britain, for the ambient temperature range 0°C to 10°C, the mean relative humidity has been shown to be 80% by Heap [71]. Using this and the empirical relationship between evaporating temperature and ambient enthalpy given in Appendix 3, graphs of evaporating temperature v ambient temperature have been plotted over the full engine speed range and are given in Figure 4.8.

These graphs show that at only 1500 rev/min is the temperature approach within the limit proposed by Carrington [38], and at a speed of 3000 rev/min it is in excess of 15K.

Besides the reduction in performance previously discussed, this high temperature approach results in an increased rate of frost formation on the evaporator, and raises the minimum ambient temperature required to prevent excessive frost formation.

The temperature approach could be reduced by increasing the physical size of the evaporator, increasing the ambient air flow rate, or by improving the heat transfer characteristics of the evaporator.

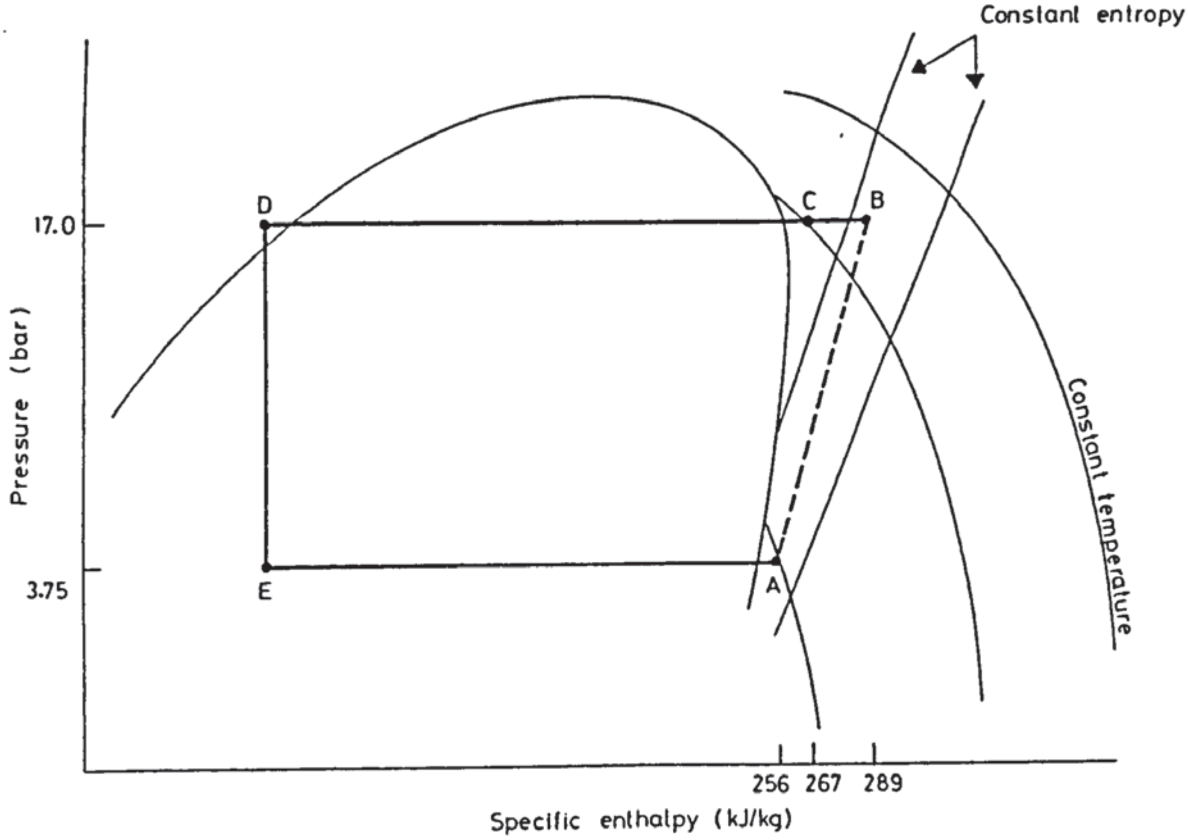


Figure 4.9 Typical Pressure v Specific Enthalpy Diagram For The Refrigeration Cycle Of The Gas Engine Driven Heat Pump System
Data Base: Enthalpy Of Saturated Liquid At 233.15K = 0 kJ/kg

Figure 4.9 has been plotted from a typical set of experimental data, where:

A is the compressor suction
B is the compressor discharge
C is the inlet to the condenser
D is outlet from the condenser
E is inlet to evaporator

From the diagram it appears that entropy decreases during the compression process (A-B). When the work of compression is calculated from these refrigerant characteristics and thermodynamic considerations, it appears to be significantly less than the absorbed power measured using strain gauges mounted on the compressor shaft (see Figure 4.10, page 75).

A similar discrepancy exists when the condenser performance (C-D) is compared with the condenser output determined from the cooling water characteristics, and because of these differences, an energy balance for the refrigerant plant could not be obtained.

The reasons for these discrepancies were not immediately apparent. However extensive tests were carried out on the compressor to trace the cause of these problems, and the results of this work are reported in Chapter 5.

The sensible heat loss in the compressor discharge line (B-C) can be clearly seen in Figure 4.9. Lagging the refrigerant pipework would considerably reduce this and would result in extra useful heat being available at the condenser.

Since the refrigerant dryness fraction at inlet to the evaporator cannot easily be measured, the expansion process (D-E) is assumed to be isenthalpic. There is some subcooling of the liquid refrigerant leaving the condenser, however, because of limitations of the system, the effects of varying the subcooling could not be explored. If the water supply temperature to the condenser were sufficiently low, increasing the level of subcooling would be advantageous. The refrigerating effect would be increased, by lowering the dryness fraction of the refrigerant entering the evaporator. This would be particularly pertinent for applications with both heating and cooling load requirements.

The characteristics of the evaporator calculated from the properties of the ambient air are not presented. Problems associated with the measurement of the air flow rate and differential temperature across the evaporator are such that data integrity cannot be ensured. Previous work on an air to air gas engine driven heat pump by the Building Research Establishment [72] highlights this factor.

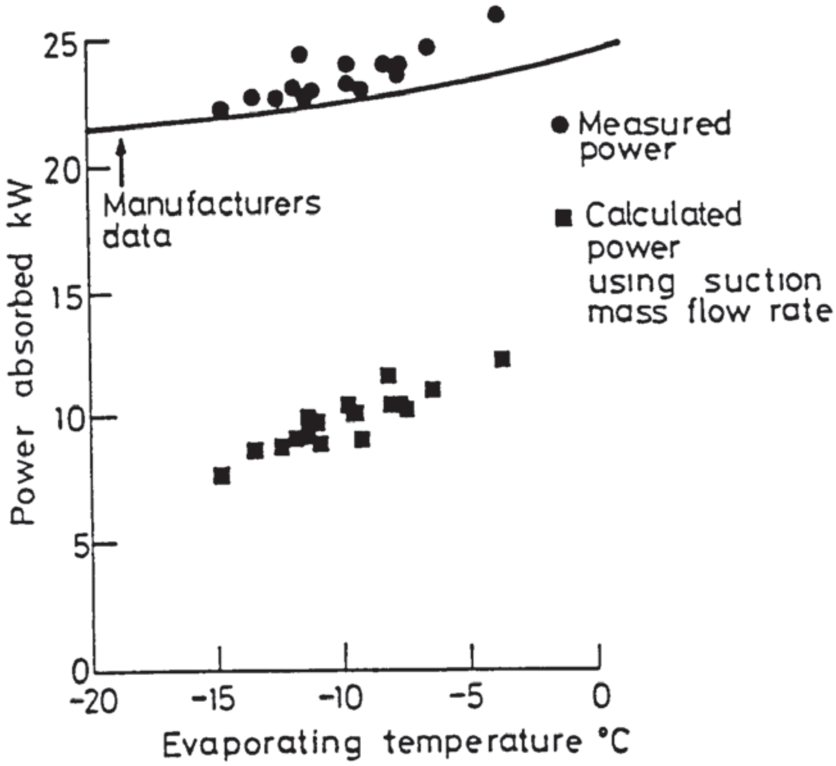


Figure 4.10 Compressor Power Absorbed v Evaporating Temperature
Compressor Speed: 3000 Rev/Min
Condensing Temperature: 45°C

The normal method used, in the refrigeration industry, to measure heat extraction at the evaporator, is to enclose the evaporator in a well insulated box, which has a very low thermal conductivity, and to operate it against electrical resistance heaters. The heat input by the electrical resistance heaters can be accurately measured, and the insulated box is designed such that losses to atmosphere are negligible. This method was impractical for the gas engine driven heat pump due to the size of the plant.

4.2.2 Engine Performance

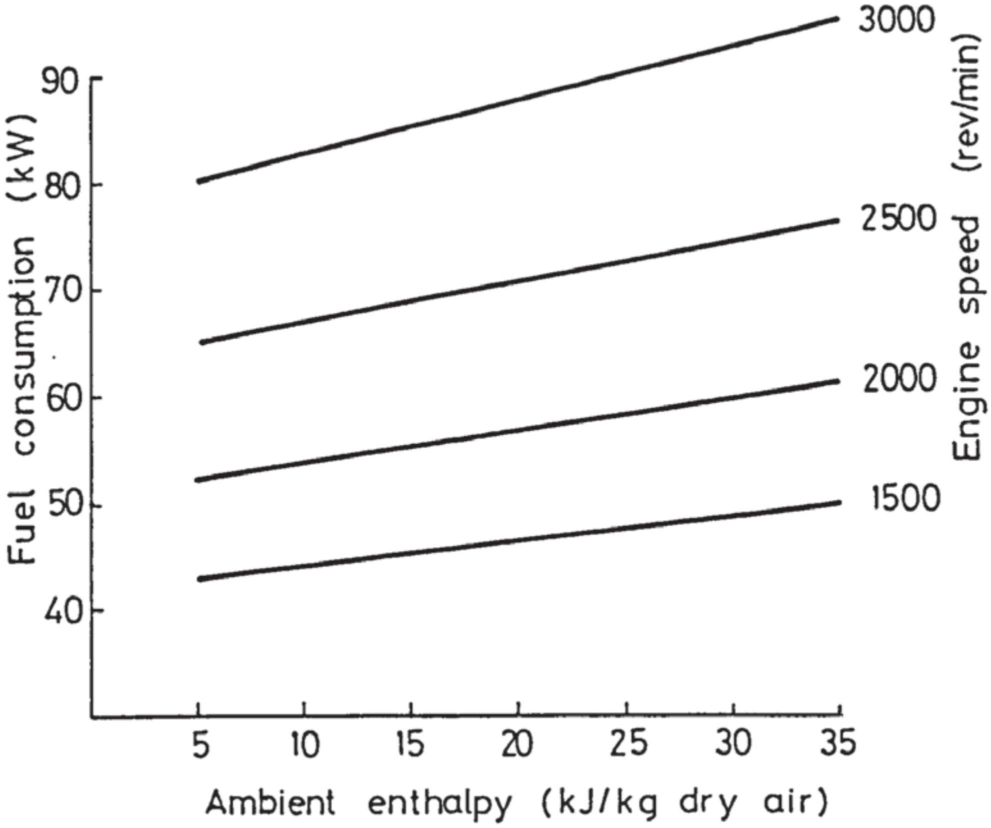


Figure 4.11 Fuel Consumption v Ambient Enthalpy

The rate of fuel usage (Figure 4.11) increases with:

a) Ambient Enthalpy. The power absorbed by the compressor increases with ambient enthalpy (Figure 4.3, page 66). Since an engine thermal efficiency (brake power/fuel consumption) is relatively constant, the fuel consumption must increase with ambient enthalpy.

b) Speed. Fuel consumption must increase with increased engine speed, as does the power.

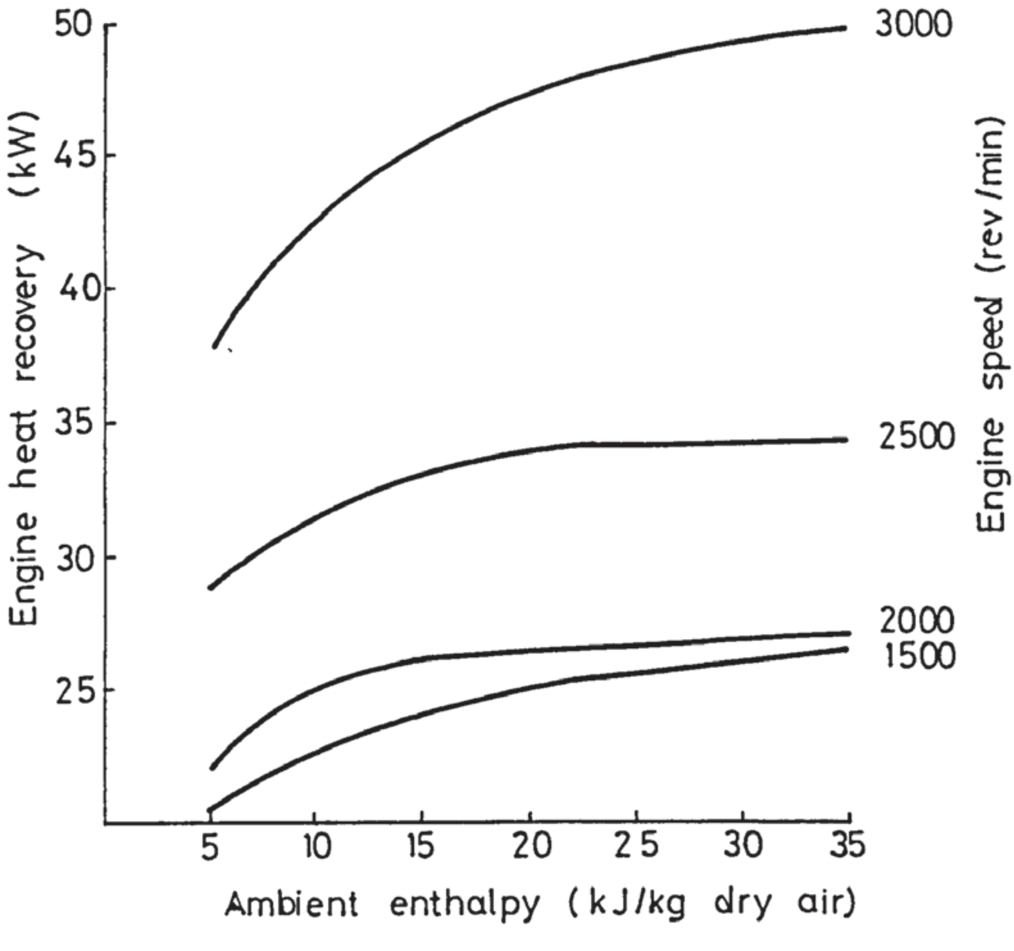


Figure 4.12 Engine Heat Recovery v Ambient Enthalpy

Figure 4.12 shows the engine heat recovery using the Serck heat recovery equipment and this is combined with Figure 4.11 and Figure 4.3 (page 66) to obtain the gas engine heat balance plotted in Figure 4.13. This engine heat balance indicates that up to 30% of the heat content of the fuel is rejected to waste. There are three possible explanations:

a) The Serck heat recovery equipment appears to be inefficient, and has a high exhaust gas temperature.

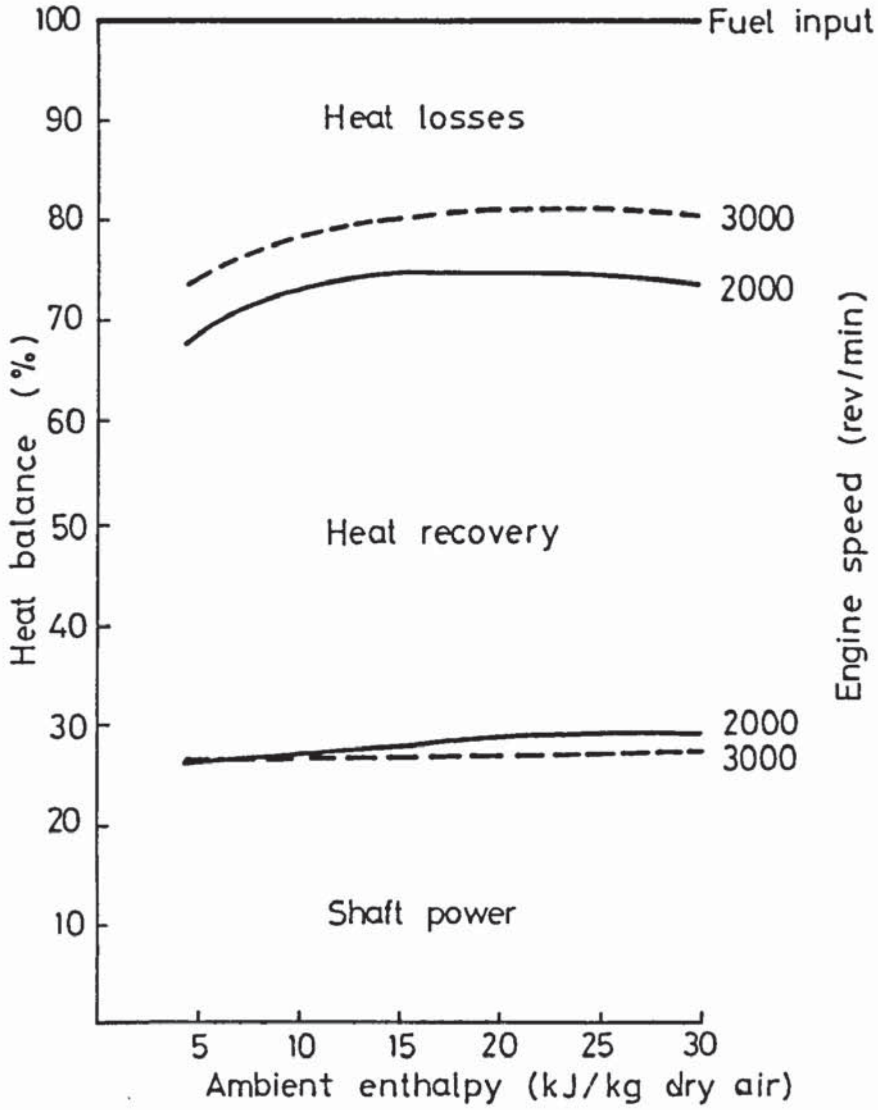


Figure 4.13 Gas Engine Heat Balance

b) The evaporator is mounted at the rear of the engine/compressor module, and air (in excess of 11,000 cfm) is drawn across the engine before it passes through the evaporator. Originally it was considered that this arrangement would enable any convected heat to be recovered at the evaporator. However, the very high air flow rate induces forced convection which considerably lowers the engine temperature, and extracts heat from the heat recovery equipment, which, like the refrigeration pipework, is unlagged.

c) The heat content of the exhaust gases quoted by the manufacturer is based on tests carried out by the British Gas Midland Research Station. Discussions with engineers at British Gas [73] indicated that the exhaust gas heat content had been computed using an ambient temperature base of 15^{o}C. A large proportion of the heat contained in the exhaust gas is latent heat, and since the supply water temperature to the heat recovery equipment is higher than the exhaust gas dew point, this latent heat cannot easily be recovered.

Because of the above considerations, and the poor reliability of the Serck heat exchanger system (see Section 8.7) it was decided to utilise Bowman engine heat recovery equipment for the production heat pumps. Results of work on the Bowman equipment are recorded in Chapter 6.

4.2.3 Overall System Performance

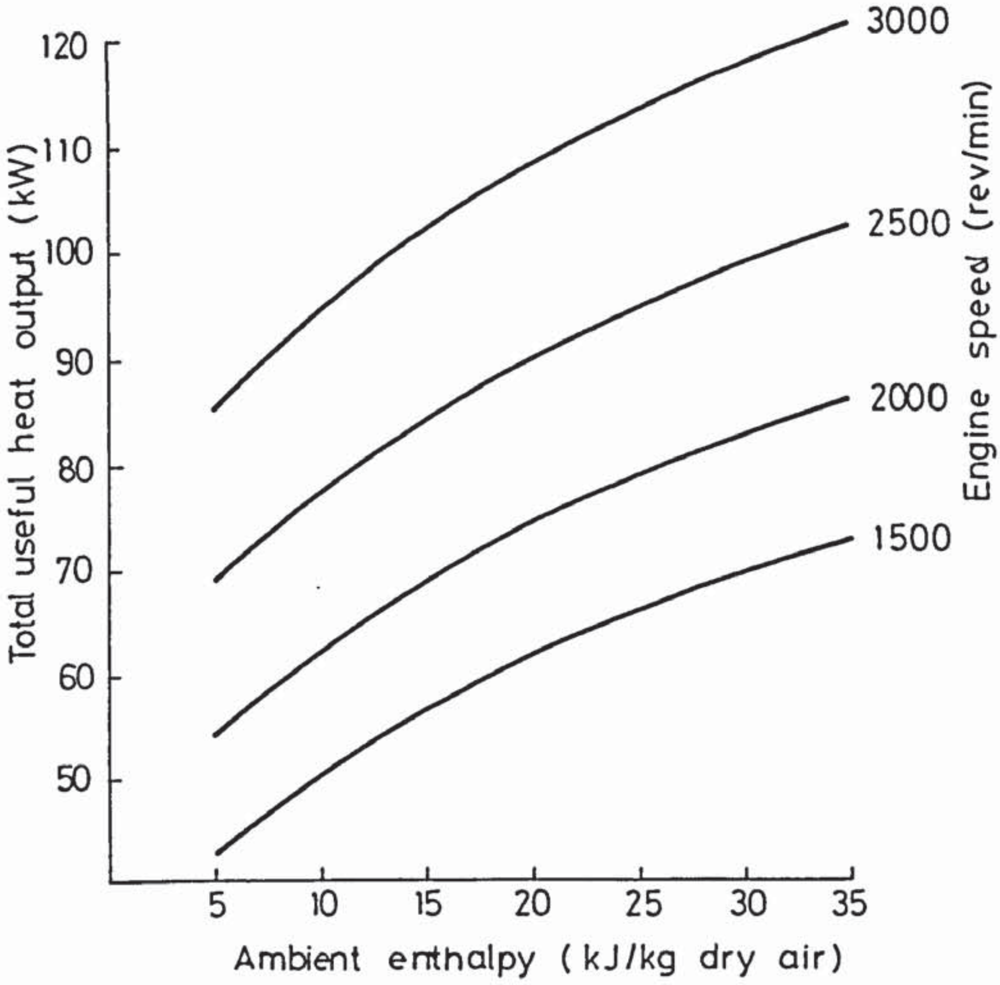


Figure 4.14 Total Useful Heat Rejected v Ambient Enthalpy

The total system output is the sum of the condenser output and the engine heat recovered. The total useful heat output was found to be lower than initially calculated from the engine and compressor manufacturers' data, because of the problems associated with heat recovery, and the low evaporating temperatures experienced.

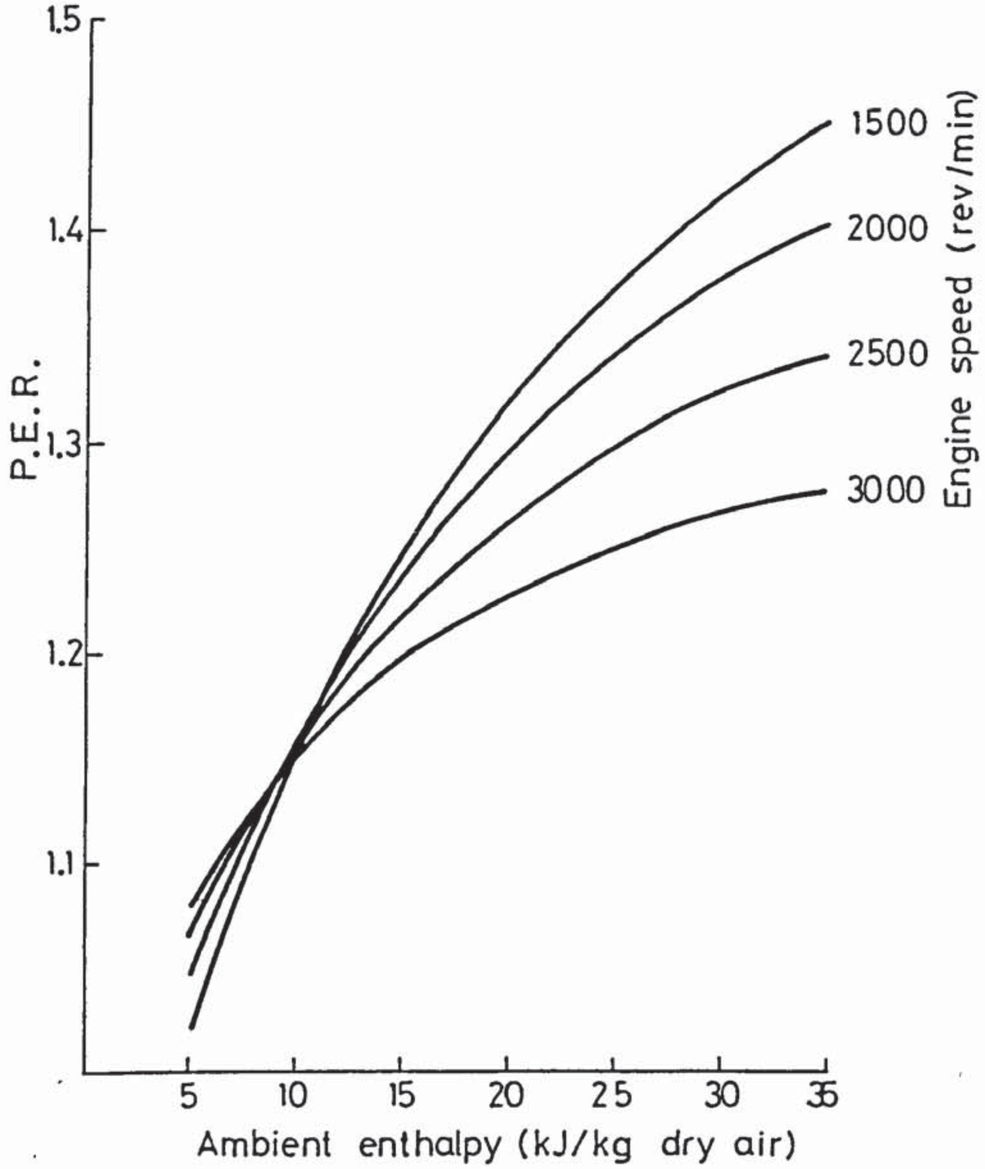


Figure 4.15 Primary Energy Ratio v Ambient Enthalpy

The primary energy ratio (P.E.R.) is the ratio of total useful heat output from the system to the energy supplied in the fuel, and in this work has been determined by combining the empirical relationships from Figures 4.11 and 4.14 contained in Appendix 3.

4.2.4 Comparison Of Steady State Performance With The Compressor Manufacturer's Published Data

Comparison of the experimental results for the system components with the manufacturer's data is essential. The manufacturers quote compressor performance as a function of evaporating and condensing temperature, so it is necessary to present the experimental data for this project in the same form.

Thus Figure 4.16 is derived from Figures 4.3, 4.6 and 4.7, and Figure 4.17 is derived from Figures 4.1, 4.6 and 4.7. For speeds less than 3000 rpm, the manufacturers state that both the compressor power absorbed, and the condenser output should be reduced pro-rata to the speed.

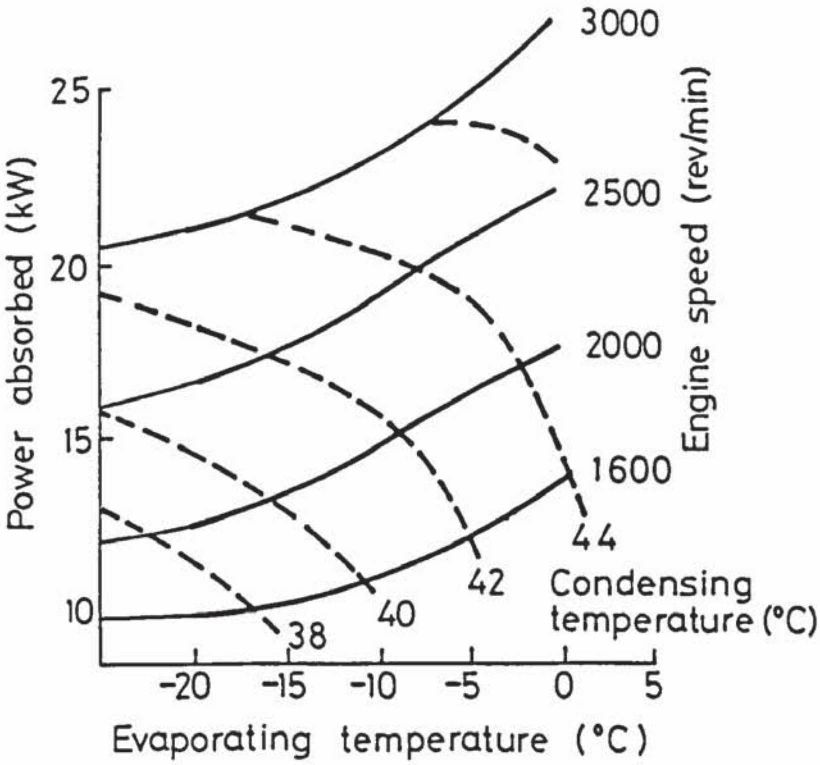


Figure 4.16 Compressor Power Absorbed, And Condensing Temperature v Evaporating Temperature and Compressor Speed

Working in the opposite direction, the values of shaft power and condenser output, from Figures 4.16 and 4.17, were increased pro-rata up to equivalent values for a speed of 3000 rpm, for the condensing temperature range shown.

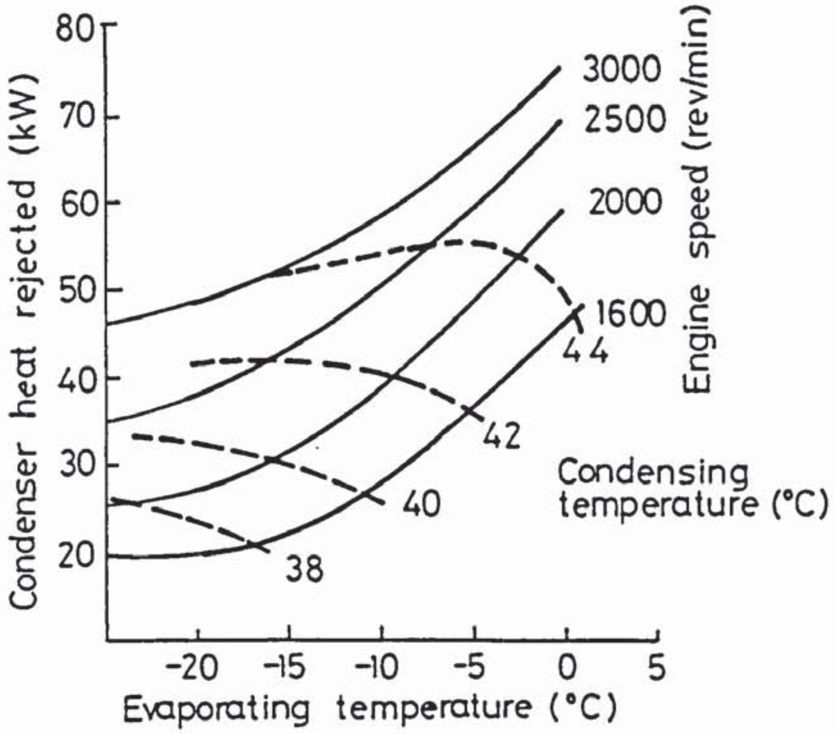


Figure 4.17 Condenser Heat Rejected And Condensing Temperature v Evaporating Temperature And Speed

The values of power absorbed and condenser output obtained in this manner were then superimposed onto graphs of the compressor manufacturer's published data (see Figures 4.18 and 4.19). A good correlation with the manufacturer's published data, for both compressor power and condenser output, has been obtained.

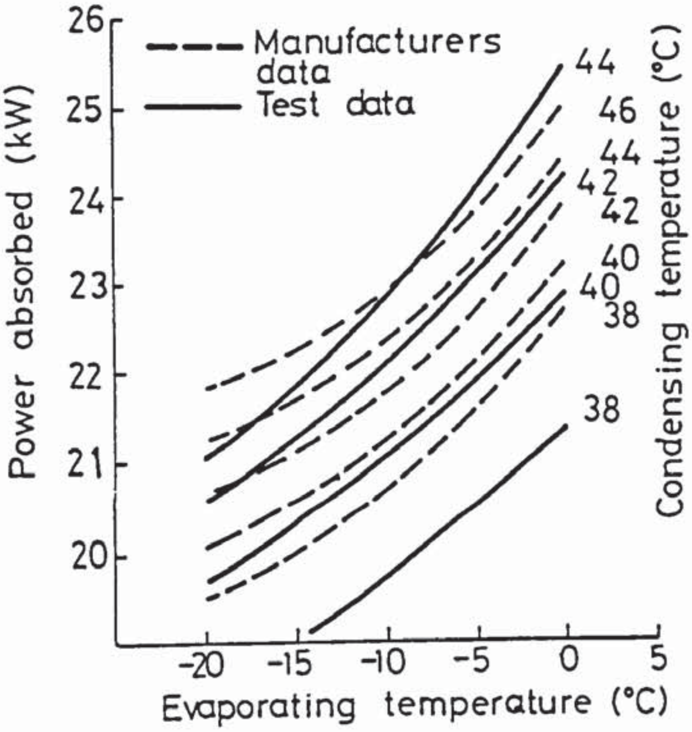


Figure 4.18 **Compressor Absorbed Power v Evaporating & Condensing Temperature**
Comparison Of Experimental & Manufacturer's Published Data
Compressor Speed: 3000 Rev/Min

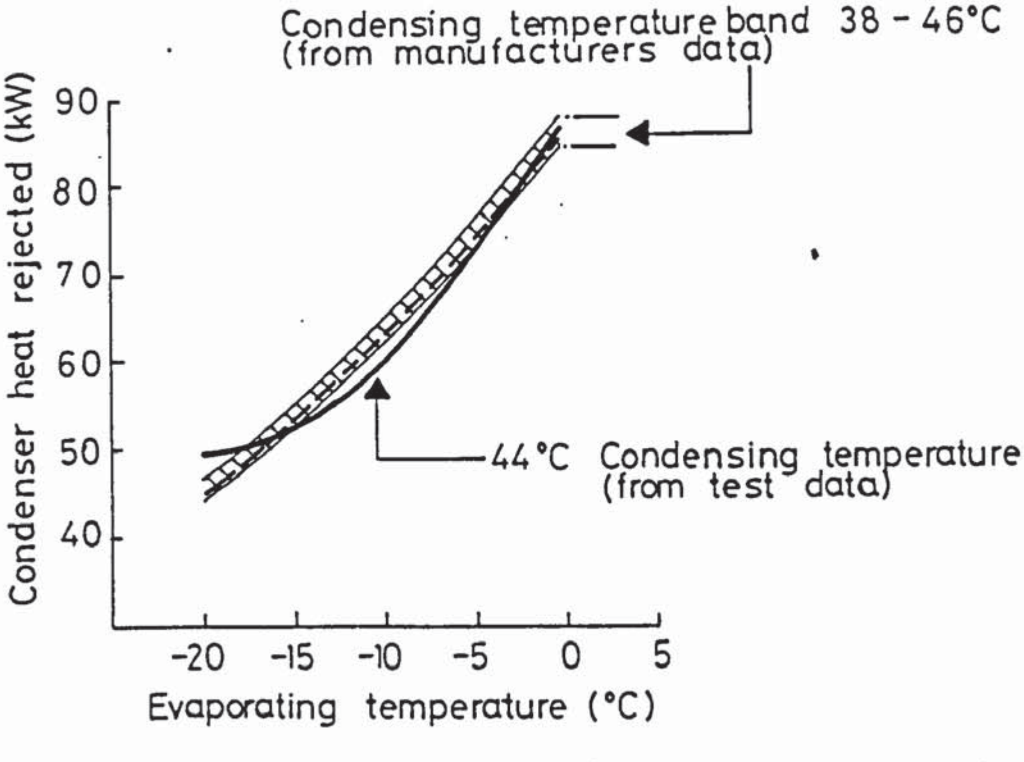


Figure 4.19 **Condenser Heat Rejected v Evaporating & Condensing Temperatures**
Comparison of Experimental & Manufacturer's Published Data
Compressor Speed: 3000 Rev/Min

The deviation between test and published data is less than 5% for the range of conditions considered.

It should be noted that due to the narrow band of condensing temperature obtained from the experimental work, only one condensing temperature (44°C) is shown in Figure 4.19. The 44°C condensing temperature was selected because test data spanned the full range of engine speed at this temperature. It appears from Figure 4.17 that maximum condenser heat rejection occurs at a speed of 2200 rev/min for the 44°C condensing temperature. The reason for this is not apparent, however it could be related to the compressor vane chatter experienced at low operating speeds.

4.2.5 Comparison Of Steady State Performance With The Engine Manufacturer's Published Data

The manufacturer's published data for engine performance is presented as a function of speed and engine loading.

To present the experimental data for this project in the same form, Figure 4.20 has been derived from Figures 4.11 and 4.3, and Figure 4.21 is derived from Figures 4.12 and 4.3. The engine load is determined as the ratio of measured power to the full load power stated by the manufacturer.

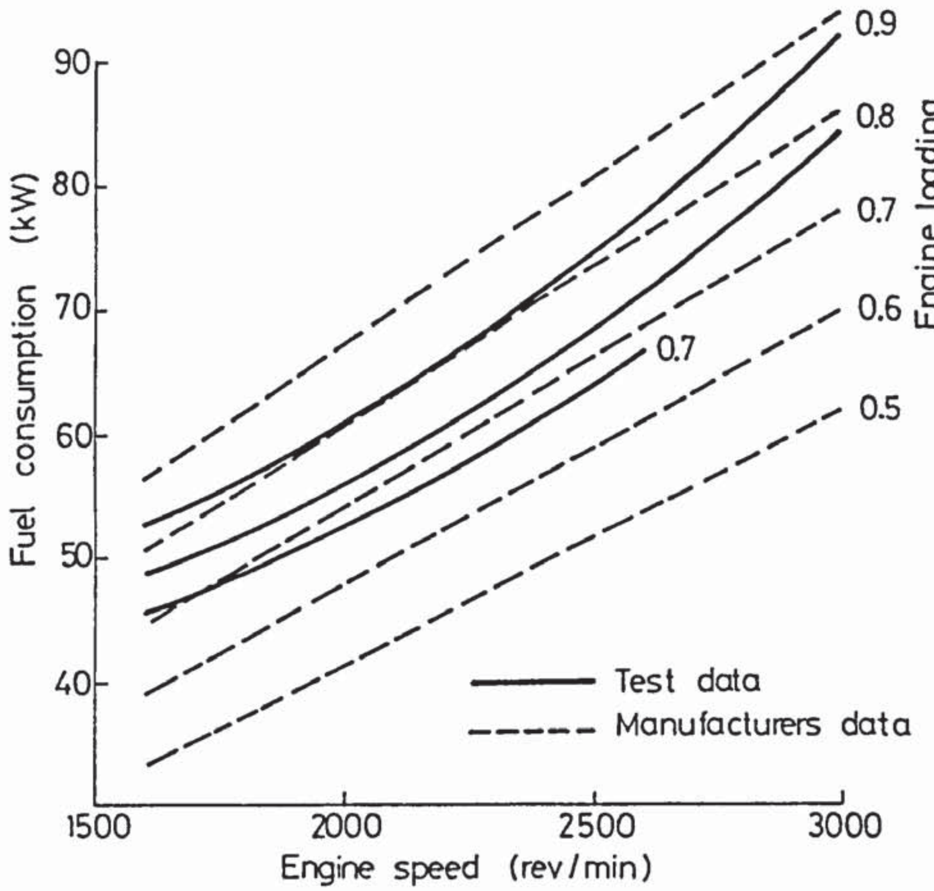


Figure 4.20 Fuel Consumption v Engine Speed & Load Comparison Of Experimental & Manufacturer's Published Data

Allowing for errors, introduced by assuming the full load power, quoted by the manufacturer, to be correct, a reasonable correlation with the manufacturer's published data for the engine fuel consumption has been obtained.

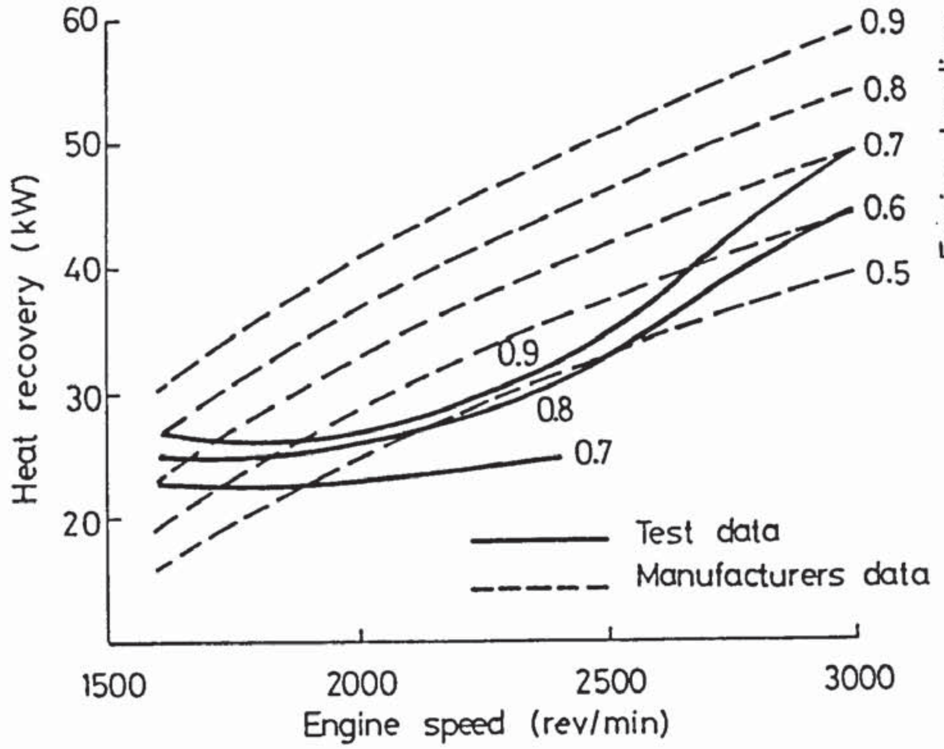


Figure 4.21 Heat Recovery v Engine Speed & Load Comparison Of Experimental & Manufacturer's Published Data

There is approximately 10% variation between the published and experimental data at a speed of 2200 rev/min, however, it is unlikely that the manufacturer's test data would produce a straight line as indicated by the published figures. Discussions with the manufacturers [74], indicated that the optimum engine speed is 2200 rev/min, which confirms this point.

The comparison of heat recovery with the manufacturer's data highlights major discrepancies, the reasons for which are discussed in Section 4.2.2.

4.3 TRANSIENT EXPERIMENTAL PROCEDURE

The formation of frost on air heated evaporators results in an increased resistance, and hence reduced air flow rate across the evaporator. The effect of this is to lower the evaporating temperature, which in turn reduces system performance for a given temperature lift.

The objective of the transient testing is to determine the effect of frost formation on system output with time, at constant operating speeds.

For the purpose of these tests only three engine speeds were considered: 1800 rev/min, 2200 rev/min and 2600 rev/min.

With the engine set at one of the above speeds the system was allowed to reach steady state conditions. A defrosting process was then initiated to clear any frost from the face of the evaporator. Readings were then obtained at five minute intervals until the condenser heat rejected fell to below 50% of the steady state value.

4.4 PRESENTATION AND DISCUSSION OF THE TRANSIENT RESULTS

Parameters on the graphs contained in this section are presented as a percentage of the steady state performance. Because of the nature of frost formation, the evaporating temperature of the system falls over a period of time, even for constant ambient conditions. This fall of evaporating temperature with time results in an increased temperature approach.

Thus:

$$\text{\% Fall of evaporating temperature} = \frac{(\text{ambient} - \text{evaporating temp})(to)}{(\text{ambient} - \text{evaporating temp})(t\ )}$$

where (to) is the initial time. (4.1)

Energy must be used to dissipate this frost, so an attempt is made to quantify the effects of frost formation on air heated evaporators, by considering a typical heating/defrosting cycle.

It should be noted that the data presented here is only applicable to the system tested, and cannot be related to other systems which may not have the same physical characteristics. However, the trends are of significance.

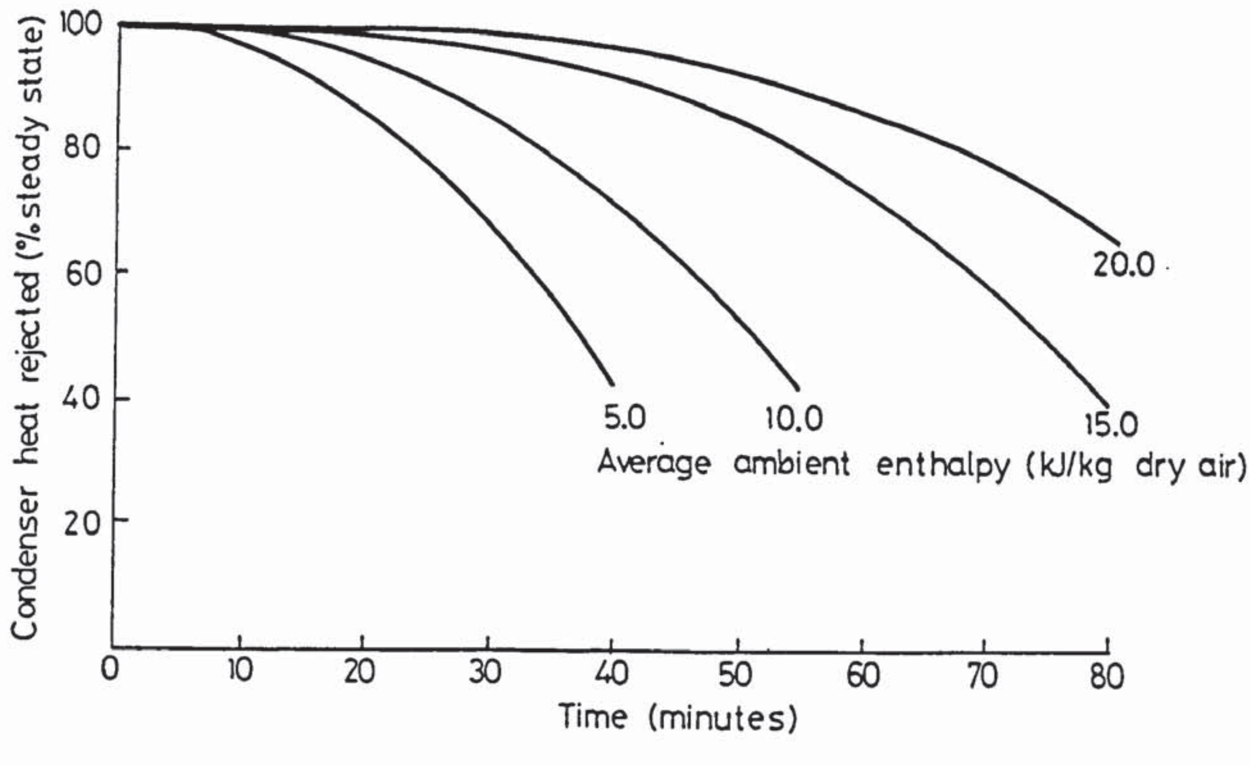


Figure 4.22 Condenser Heat Rejected v Time (Compressor Speed 2200 Rev/Min)

Figure 4.22 shows the fall of condenser heat rejected with time for a range of ambient enthalpy conditions, at a speed of 2200 rev/min.

It can be seen that the condenser heat rejected has a higher rate of decline at low ambient conditions. This is because the rate of frost formation is higher at these lower ambient conditions.

Similar trends occur when a range of speeds are considered, as can be seen from Figure 4.23.

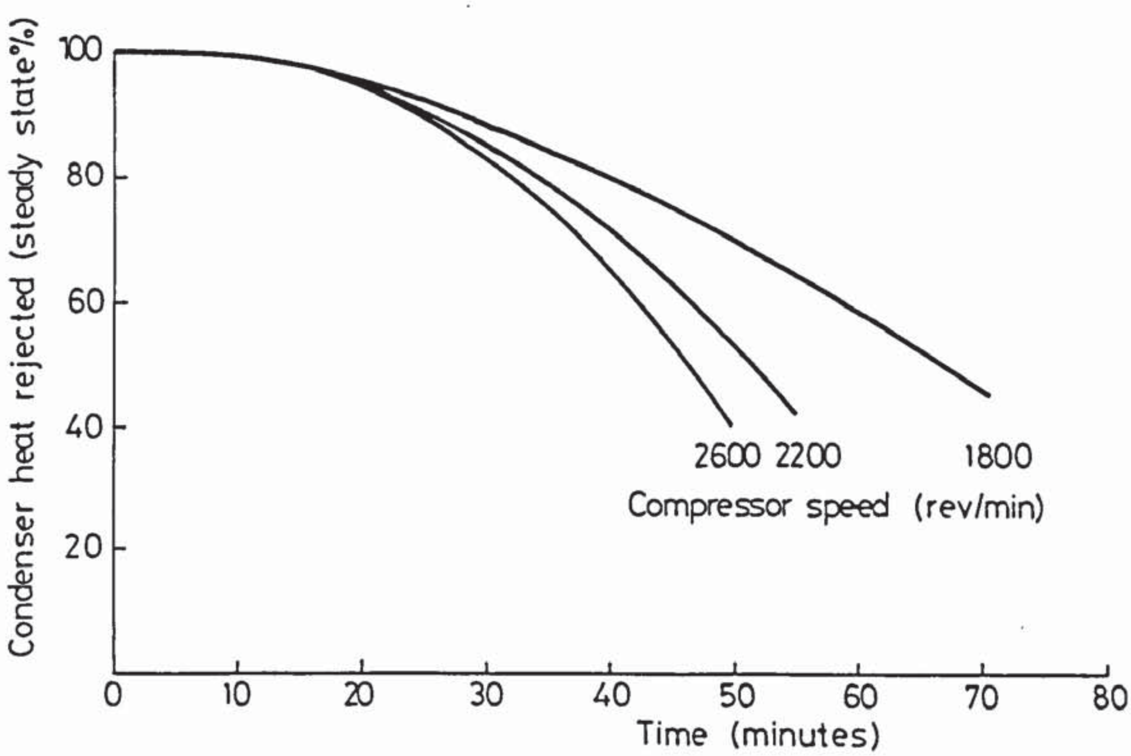


Figure 4.23 Condenser Heat Rejected v Time (Ambient Enthalpy 10 kJ/kg dry air)

As frost builds up on the face of the evaporator the air flow rate is progressively suppressed and there is less heat transfer. Since the compressor volumetric flow rate is relatively constant for a given speed, the expansion valve closes in an attempt to maintain equilibrium, thereby lowering the evaporating pressure and temperature. This results in a reduction of the refrigerant density and hence system mass flow rate. This fall in evaporating temperature is apparent from Figure 4.24.

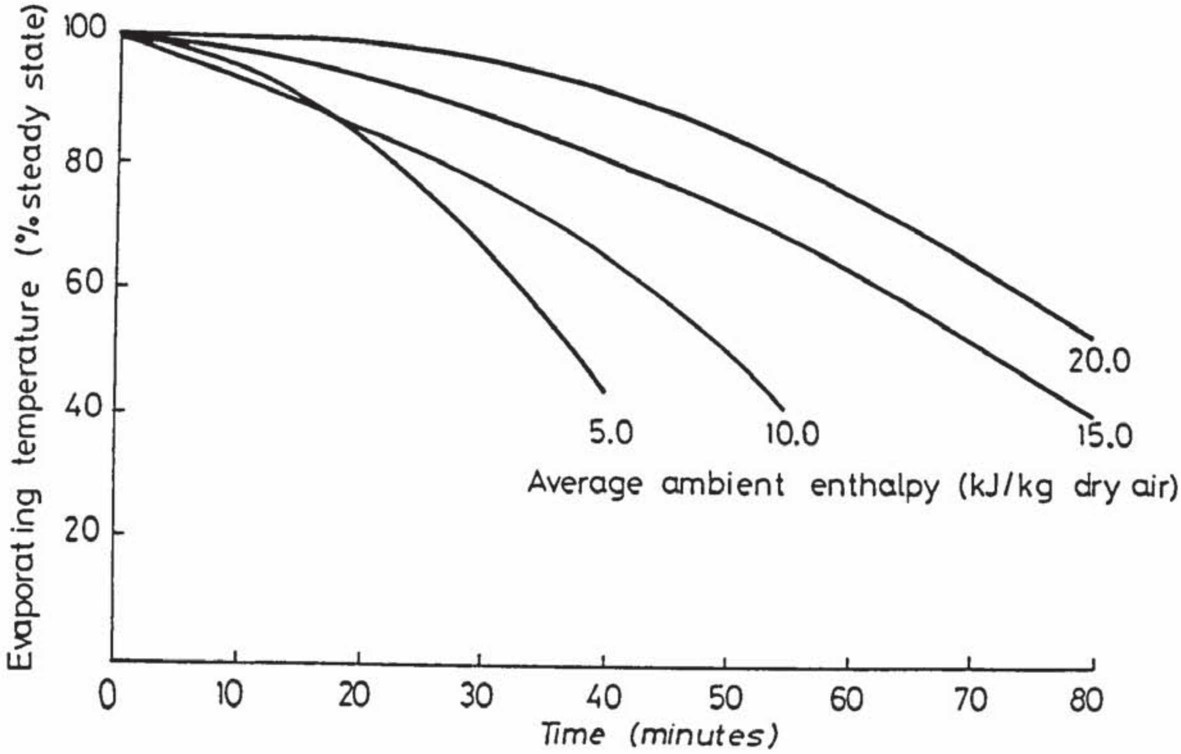


Figure 4.24 Evaporating Temperature v Time (Compressor Speed 2200 Rev/Min)

A consequence of this is that the expansion valve is unable to regulate the refrigerant superheat and so hunts very badly in an attempt to stabilize the compressor suction conditions.

As the compressor speed increases (Figure 4.25) the volumetric flow rate of the refrigerant increases, with similar results.

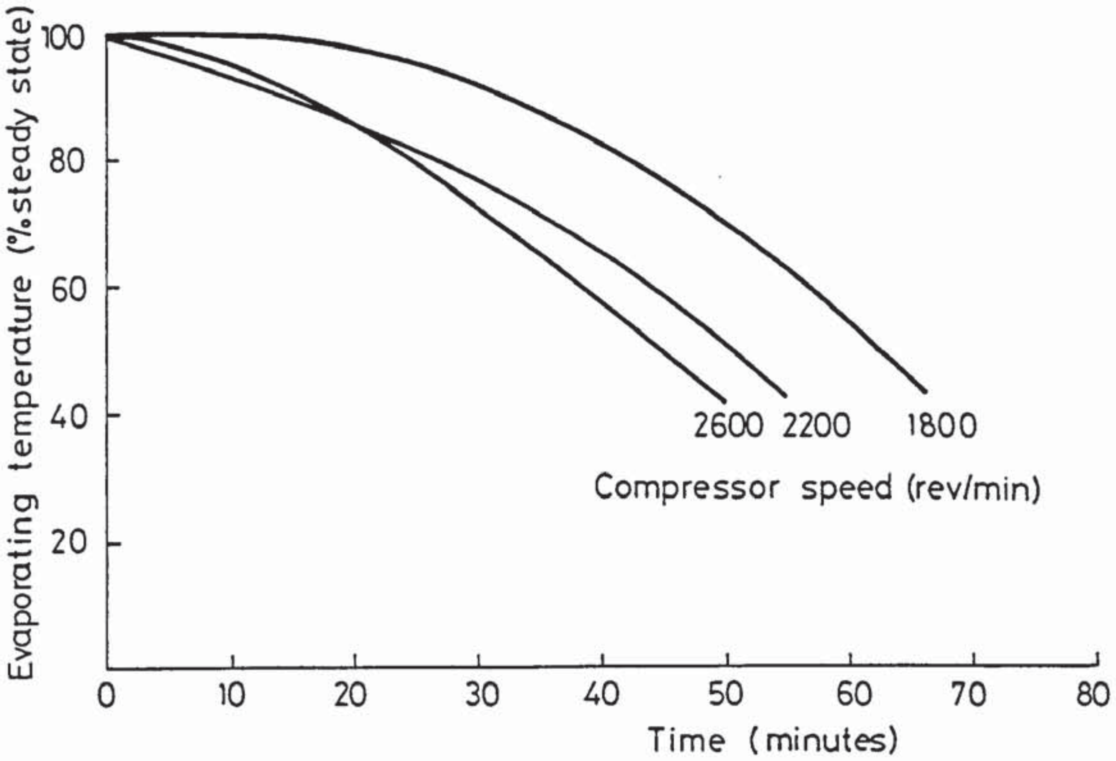


Figure 4.25 Evaporating Temperature v Time (Ambient Enthalpy 10 kJ/kg dry air)

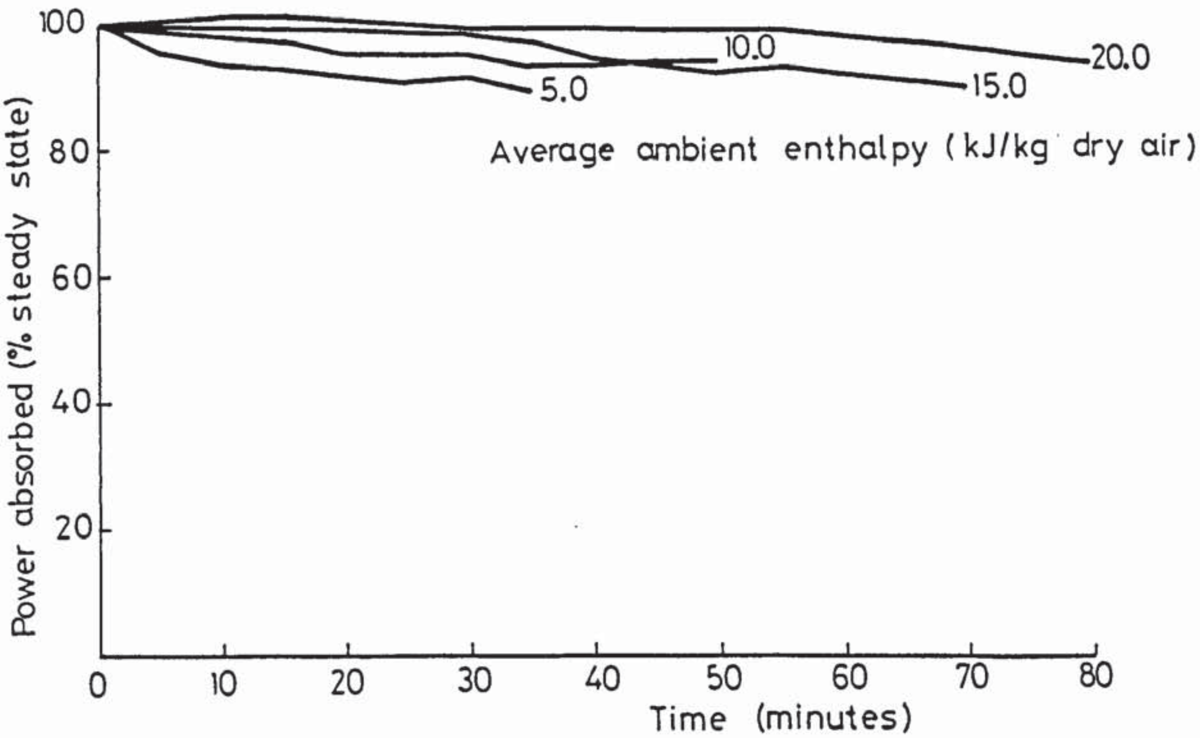


Figure 4.26 Power Absorbed v Time (Compressor Speed 2200 Rev/Min)

As frost builds up on the face of the evaporator the system C.O.P. falls, since the power absorbed by the compressor is largely unaffected by the frost formation (Figure 4.26).

This would also be true for an electric heat pump. However, in the case of a gas engine heat pump, since the power absorbed remains relatively constant, the engine heat recovery is unaffected by frost formation. The total useful heat output from the system will not be substantially reduced since the reduced heat output from the condenser is only a fraction of the total heat output. This is a major advantage for the gas engine driven heat pump when compared with an electric heat pump.

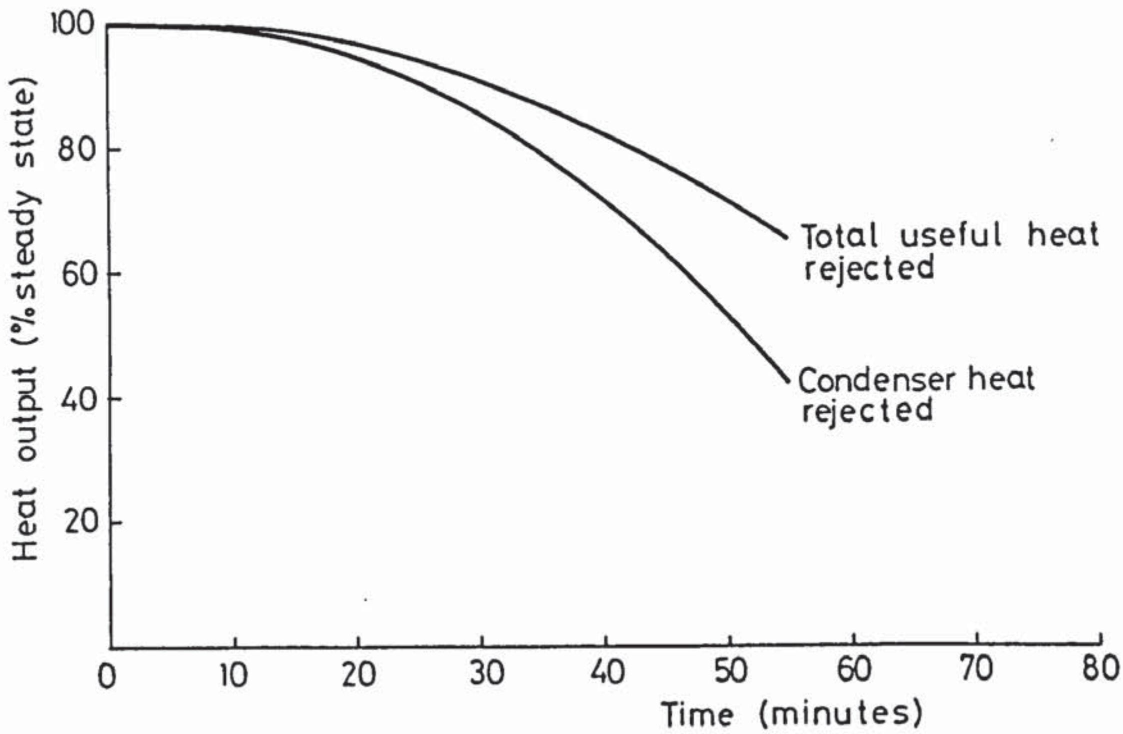


Figure 4.27 Comparison Of Reduced Total Useful Heat & Condenser Heat Rejected With Time (Compressor Speed 2200 Rev/Min) (Ambient Enthalpy 10 kJ/kg dry air)

It is essential to remove frost build up periodically, otherwise heat transfer at the evaporator would eventually be suppressed. The evaporating pressure would fall below a minimum, at which time the plant would shut down because of actuation of the low pressure protection device.

Conventional defrosting techniques use energy to dissipate the frost, and this results in a further reduction of energy efficiency.

For the prototype gas engine driven heat pump, hot vapour defrosting was used. Because of compressor related problems this was limited to a maximum of 60 seconds. Once the refrigerant has condensed in the evaporator, the compressor acts as a liquid pump, drastically reducing the pressure differential between the compressor suction and the oil separator. Since a pressure differential is necessary for oil return to the compressor, lubrication is inhibited. If allowed to continue for an extended period, this would ultimately result in damage to the seals, bearings and vanes of the compressor.

This hot vapour defrosting has been adopted for the production gas engine driven heat pumps, and a timer relay initiates and terminates defrosting. This timer is set such that the total cycle time is 46 minutes with a 60 second defrosting process. An outdoor thermostat is used to inhibit the defrosting process at all ambient temperatures above $10^{\circ}C$.

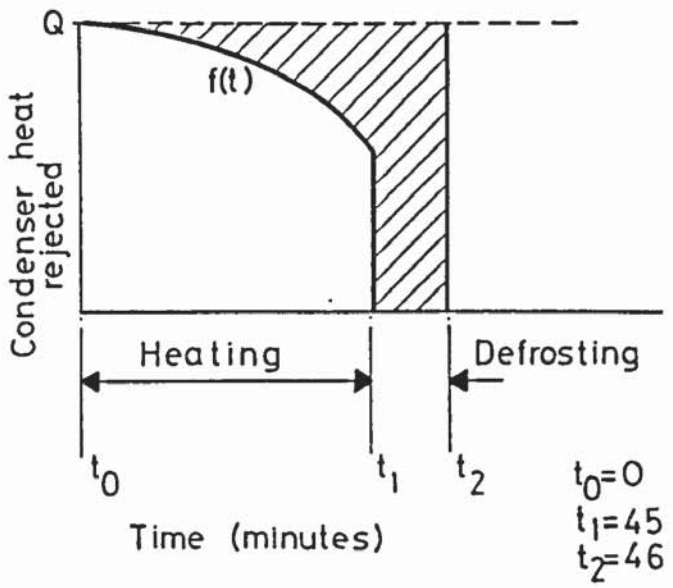


Figure 4.28 Representation Of Transient Heat Pump Output

Figure 4.28 represents the heat output of a heat pump utilising periodic hot vapour defrosting, where the shaded area is the heat lost due to frost formation and its subsequent removal.

During the heating process:

$$\text{useful heat} = \int_{t_0}^{t_1} f(t)\,dt$$

Thus:

$$\%\ \text{heat lost} = 100x(Q(t_2-t_0)-\int_{t_0}^{t_1} f(t)dt)/(Q(t_2-t_0)) \qquad (4.2)$$

The empirical relationship for the transient results at a speed of 2200 rev/min is:

$$f(t) = 100+(-0.226+0.0653h-2.28x10^{-3}h^2)t + (-0.0582+4.795x10^{-3}h-1.072x10^{-4}h^2)t^2 \qquad (4.3)$$

where t is the time (minutes)
and h is the ambient enthalpy (kJ/kg dry air)

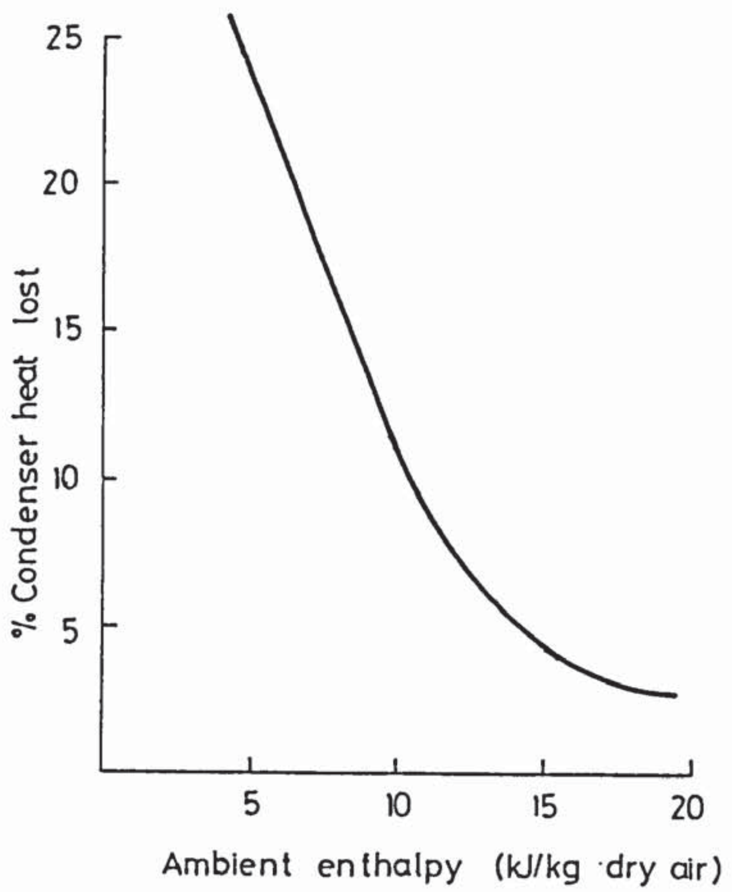


Figure 4.29 Heat Lost Due To Frost Formation And Its Subsequent Removal, For A 46 Min. Cycle Time. (Compressor Speed 2200 Rev/Min)

Figure 4.29 is plotted from equations 4.2 and 4.3 for the cycle time used for the production gas engine heat pumps. Although Heap [6] reports that frost formation and its subsequent removal results in a loss of between 5% and 10% of the total useful heat output during a heating season, it is more important to consider the mid-winter conditions. This is because a building heat load is at its highest and heat pump efficiency is at its lowest. For the prototype gas engine heat pump at a speed of 2200 rev/min with an ambient enthalpy of 5kJ/kg dry air, frosting and its subsequent removal reduces the condenser heat rejected by 24.5% and the total useful heat output by

14.5%. This results in a P.E.R. of 1.03 compared with the steady state P.E.R. of 1.21. For a similar electric heat pump the P.E.R. would be reduced to 0.54 at an ambient enthalpy of 5 kJ/kg, assuming 30% electricity generating efficiency, which is considerably less than equivalent gas boiler efficiencies.

Although the frost formation could be reduced by increasing the volumetric air flow rate and the physical size of the evaporator, and by improving the heat transfer characteristics, it could not be completely suppressed. It is the major disadvantage of using ambient air as the heat source.

4.5 SUMMARY

The steady state performance of the prototype gas engine driven heat pump is comparable with the manufacturer's published data. However the system performance is considerably lower than anticipated for the following reasons:

a) The evaporator is too small. This combined with insufficient volumetric air flow results in a large temperature approach between the evaporator and the ambient air, and results in a loss of heat available at the condenser, and a reduction in system C.O.P.

b) Up to 30% of the energy in the fuel is rejected to waste. Some of this can be attributed to inefficiencies in the heat recovery equipment highlighted by the high exhaust gas temperature. In addition, the location of the evaporator results in excessive forced convection from the unit.

c) The engine heat recovery quoted by the engine manufacturer includes the latent heat content of the exhaust gases, which is not recovered by the heat recovery equipment fitted to the prototype gas engine driven heat pump.

The transient tests show a marked fall off in heat output with time due to frost build up. This frost formation and its subsequent removal reduces the condenser heat rejected by up to 25%, and the total useful heat output by up to 14.5%, and is the major disadvantage of using ambient air as a heat source.

CHAPTER FIVE

COMPRESSOR INVESTIGATION

5.1 INTRODUCTION

When an energy balance was attempted, based on the refrigerant properties and thermodynamic considerations for the heat pump cycle, in Chapter Four, a discrepancy was observed. The measured heat to the condenser tube side (water-glycol mixture) was considerably higher than that from the shell side (refrigerant). Further, the power absorbed measured using strain gauges mounted on the shaft coupling the engine and compressor was much greater than the thermodynamically calculated work of compression. Figure 5.1 shows a typical heat pump cycle (ACGH) plotted on a pressure enthalpy diagram assuming 75% isentropic compression efficiency. A-C signifies the compression process of this cycle. A rotary vane compression process is superimposed on this diagram (process A-F), based on steady state experimental data for compressor suction and discharge conditions, and it appears that the entropy decreases during this compression process.
Hughes et. al. [54,58] observed similar effects using refrigerant R12 with this type of compressor. They concluded that the discrepancy was due to the presence of oil in the system, which is necessary for lubrication. In effect, this explanation suggests that the refrigerant at the compressor suction is not superheated but is in fact a liquid-vapour mixture, as indicated by point J in Figure 1.

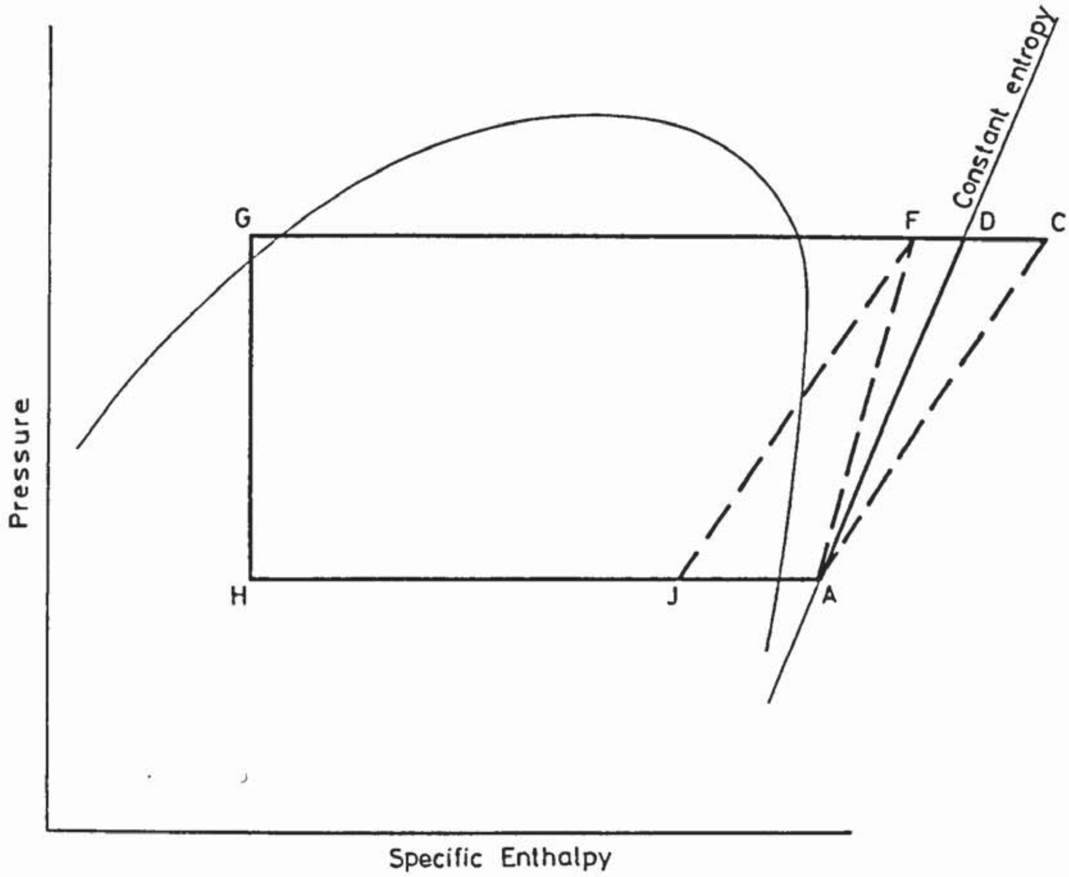


Figure 5.1 Typical Heat Pump Cycle

This can be attained by working from point F and deducting the directly measured compressor power.

Although refrigerant R12 is highly missible with lubricating oils, Downing [75] indicates that R22 is not. In addition, an oil separator was located in the compressor discharge line which minimised oil migration to the evaporator. Hence the explanation of Hughes et. al. [54,58] does not necessarily apply to this gas engine driven heat pump application.

The significant difference between reciprocating and A.G.R. rotary compressors is the injection of liquid refrigerant into the compression chamber of the rotary compressor midway through the compression process, for cooling purposes (see Figure 5.2). This results in a specific enthalpy change between compressor suction and discharge which is less than isentropic, and the refrigerant mass flow rate at compressor discharge is greater than at suction.

Because of the apparent discrepancy between measured power, and the power determined from the refrigerant properties and thermodynamic considerations, a detailed study of the compressor was initiated. The objectives were to investigate the effects of liquid injection on compressor performance and to compare the performance with:

a) a rotary compressor without liquid injection

b) a comparable reciprocating compressor.

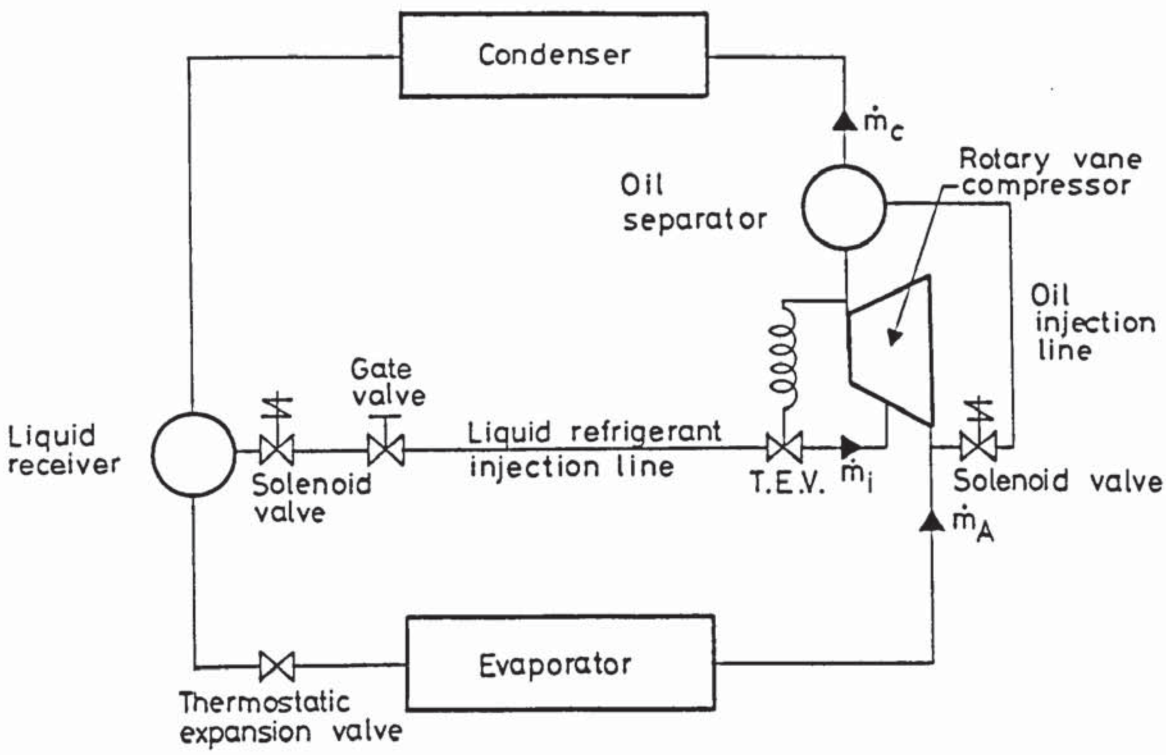


Figure 5.2 Heat Pump Schematic Diagram

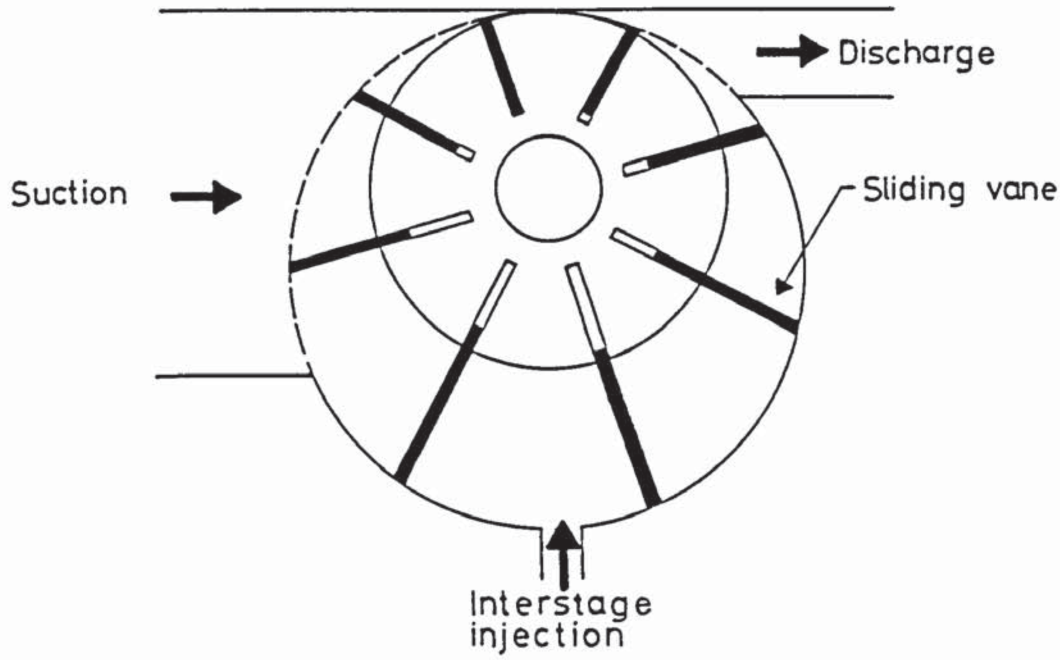


Figure 5.3 Section Through The Compressor Showing Liquid Injection

5.2 PROPOSED MECHANISM

The injection of liquid refrigerant into the compression chamber reduces the refrigerant discharge temperature as it evaporates.

Figure 5.4 shows an assumed compression process for a rotary sliding vane compressor with liquid refrigerant injection (ABEF). This process is similar to that of a two stage compression machine. B is the point of liquid injection and B-E indicates the evaporation of the liquid refrigerant, which is necessary before the compression process can continue.

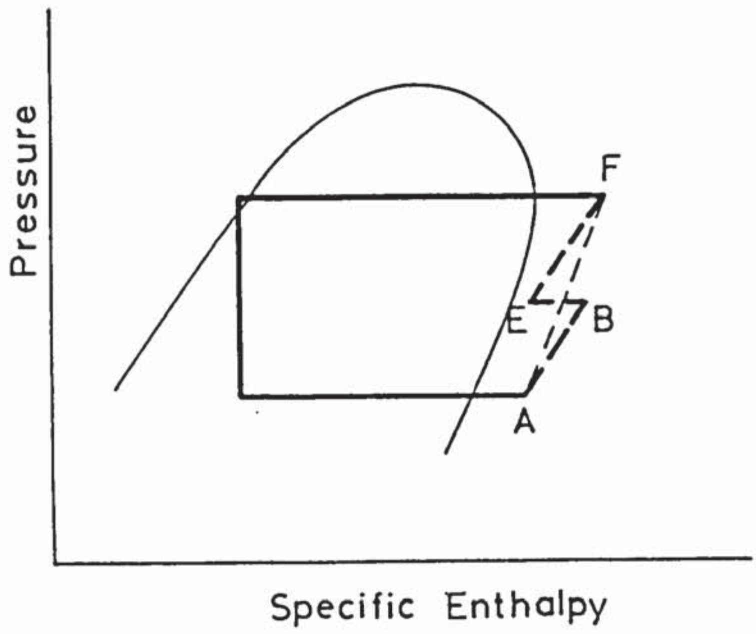


Figure 5.4 Assumed Compression Process For A Rotary Sliding Vane Compressor With Liquid Refrigerant Injection

5.3 MEASUREMENT TECHNIQUES AND EXPERIMENTAL PROCEDURES

The following parameters were measured using the instrumentation discussed in Section 3.7.

a) Compressor shaft power

b) Compressor suction and discharge temperature

c) Compressor suction and discharge pressure

The refrigerant mass flow rate at compressor suction was determined from the compressor swept volume, suction conditions, and the volumetric efficiency as measured by Woolas [70]. (The method used by Woolas [70] to determine the compressor volumetric efficiency is outlined in BS 3122 part 1 method D [76]).

In the normal mode of operation (i.e. with liquid injection) the above parameters were recorded for the engine speed range 2000 to 2500 rev/min. An oil cooler was then fitted into the oil return line between the oil separator and the compressor to enhance its lubricating

properties when liquid injection was suppressed. The tests were repeated to observe if oil cooling had any measurable effects on the compressor performance.
Liquid injection to the compressor was suppressed by electrically isolating the solenoid valve and closing the gate valve in the injection line. Results were then obtained for the same engine speed range.

5.4 PRESENTATION AND DISCUSSION OF RESULTS

A typical set of experimental results for the compressor operating with liquid injection is shown plotted on a pressure-enthalpy diagram for refrigerant R22 in Figure 5.5 (see Appendix 3 for tabulated experimental data). It can be seen that the change in enthalpy between compressor suction and discharge is less than for an isentropic process.
The introduction of the oil cooler had negligible effects on the measured parameters in the liquid injection mode of operation. When liquid injection was suppressed, the power absorbed, calculated from the refrigerant properties was comparable with the measured shaft power and the manufacturer's published data. In this mode of operation the specific enthalpy change was considerably greater than for an isentropic process, and the compressor isentropic efficiency was of the order of 70%.
The power absorbed based on the refrigerant properties is a function of both specific enthalpy and the refrigerant mass flow rate. Since liquid injection results in a higher mass flow rate at compressor discharge compared with suction, and because the power absorbed in both modes

of operation is comparable, the specific enthalpy at discharge must be lower for operation with liquid injection.

It is shown in Section 5.5 that with a 16 to 23% increase in mass flow rate at the compressor discharge the derived power is comparable with the measured power. However, because of the high density of liquid refrigerant R22 compared with vaporised R22 this is equivalent to only 0.24% liquid injection by volume.

The only penalty this liquid injection introduces into the system performance is the small amount of latent heat not absorbed at the evaporator by the interstage liquid circulating between the condenser and the compressor.

The latest information received from the compressor manufacturer indicates that compressor discharge temperatures of up to 110°C can be tolerated, provided oil cooling below 80°C is maintained, thereby eliminating the need for liquid injection. However, during the course of these experiments, when liquid injection was suppressed, heat degradation of the vane tips was experienced, causing lateral expansion of the vanes. This resulted in the inability of the vanes to extend after passing the discharge port, with the subsequent loss of compression. It is therefore considered unwise to operate these machines for extended periods without liquid injection.

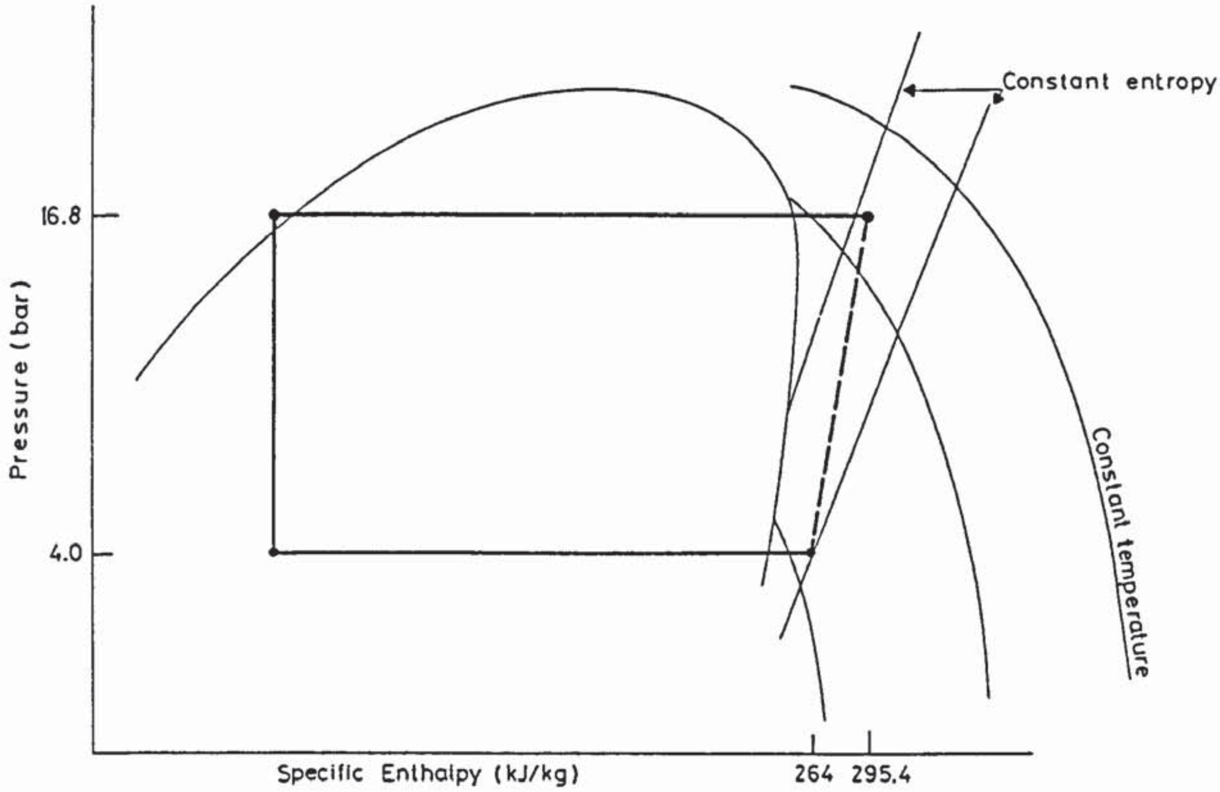


Figure 5.5 Heat Pump Cycle With Liquid Injection
Data Base: Enthalpy Of Saturated Liquid At 233.15K = 0 kJ/kg

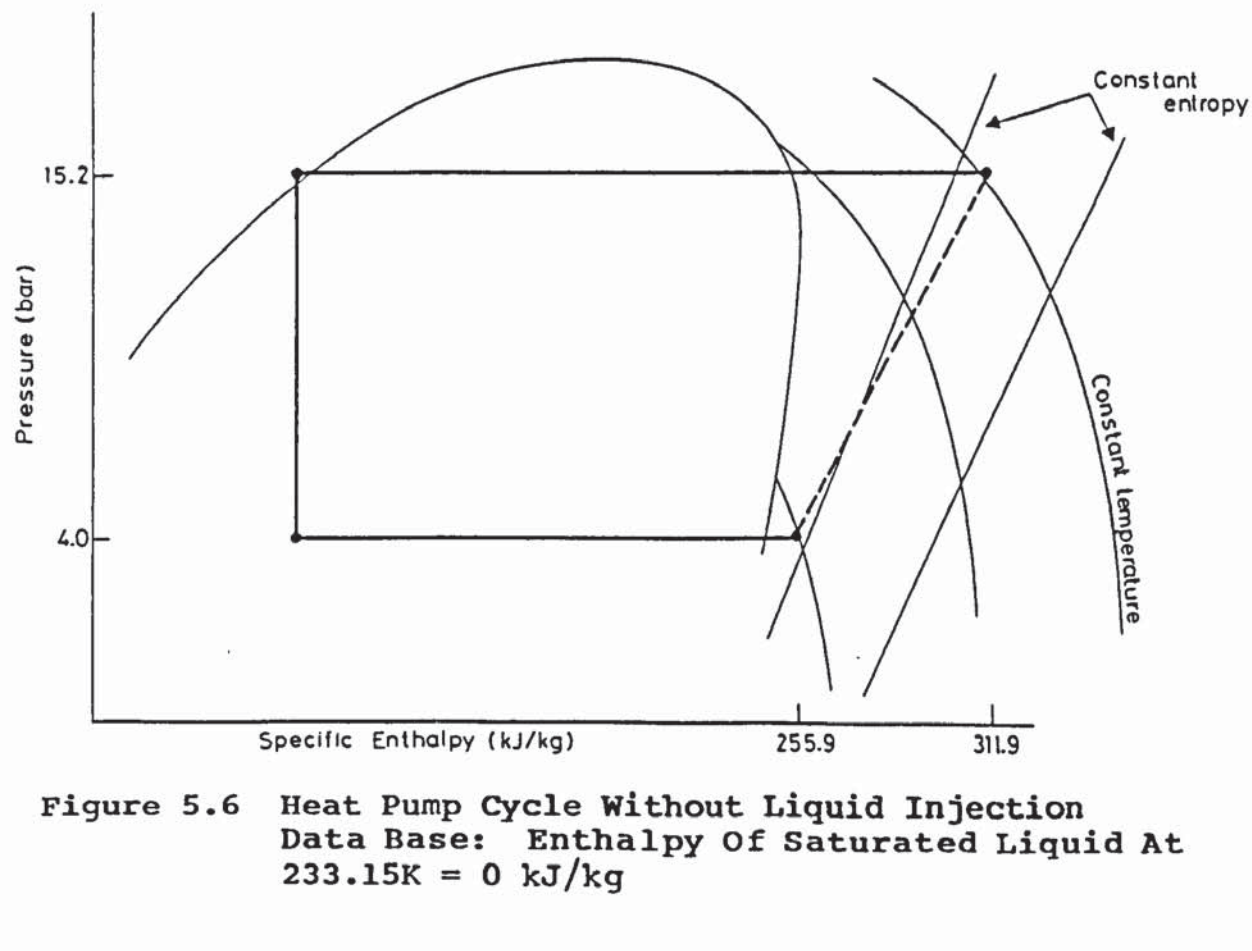


Figure 5.6 Heat Pump Cycle Without Liquid Injection
Data Base: Enthalpy Of Saturated Liquid At 233.15K = 0 kJ/kg

5.5 CALCULATED EFFECT OF LIQUID REFRIGERANT INJECTION ON COMPRESSOR MASS FLOW RATE

a) For The Compression Process:

Power Input = Increase in Enthalpy.

$$W = \dot{m}_c h_c - \dot{m}_A h_A - \dot{m}_i h_i \qquad (5.1)$$

Where:

W = power input
m_A = mass flow at compressor suction
m_c = mass flow rate at compressor discharge
m_i = interstage injection mass flow
h_A = specific enthalpy at compressor suction
h_c = specific enthalpy at compressor discharge
h_i = specific enthalpy of injected liquid

Thus transposing (5.1) we have:

$$W = \dot{m}_c(h_c - h_i) - \dot{m}_A(h_A - h_i) \qquad (5.2)$$

Appendix 3 Table 3.6.4 gives a typical set of experimental results which are used throughout the following calculations.

The mass flow rate at compressor suction is:

$\dot{m}_A$ **= 0.267 kg/s**

With liquid injection, but assuming equal mass flow at compressor suction and discharge:

W **= 10.403 kW.**

The increase in mass flow rate at compressor discharge required to give the measured power of 22.8 kW:

$\dot{m}_c$ **= 0.324 kg/s**

$\dot{m}_i$ **= 0.06 kg/s** (of the order of 23% increase)

This is less than 0.24% increase in the 0.0212 m^3/s volumetric flow rate at the compressor suction.

b) For The Condensing Process

Heat Rejected to the Cooling Water = $m_c(h_c - h_i)$ (5.3)

Thus:

$\dot{m}_c = 0.311$ **kg/s**

and:

$\dot{m}_i = 0.044$ **kg/s** (of the order of 16% increase)

The $\dot{m}_c$ value calculated in a) is greater than the value of $\dot{m}_c$ calculated in b) which is to be expected, since the condenser and the compression process are unlikely to be adiabatic. The actual value for liquid injection will lie somewhere between these two.

5.6 COMPARISON OF A ROTARY SLIDING VANE COMPRESSOR WITH A RECIPROCATING MACHINE

It has been suggested by Marquand [77] that the coefficient of performance with a rotary sliding vane compressor is less than that with a reciprocating machine. By considering the manufacturer's published data for the rotary compressor, and a typical reciprocating compressor (Figure 5.7) it can be seen that, at the maximum compressor operating speed, the condenser outputs are comparable, but the power absorbed is in fact greater for the rotary machine. However, for heat pump space heating applications the maximum heat output from the condenser is only required at low ambient temperatures. As the ambient temperature rises the building heat load for a space heating application falls, but, the heat pump output increases, resulting in the necessity to off-load the compressor.

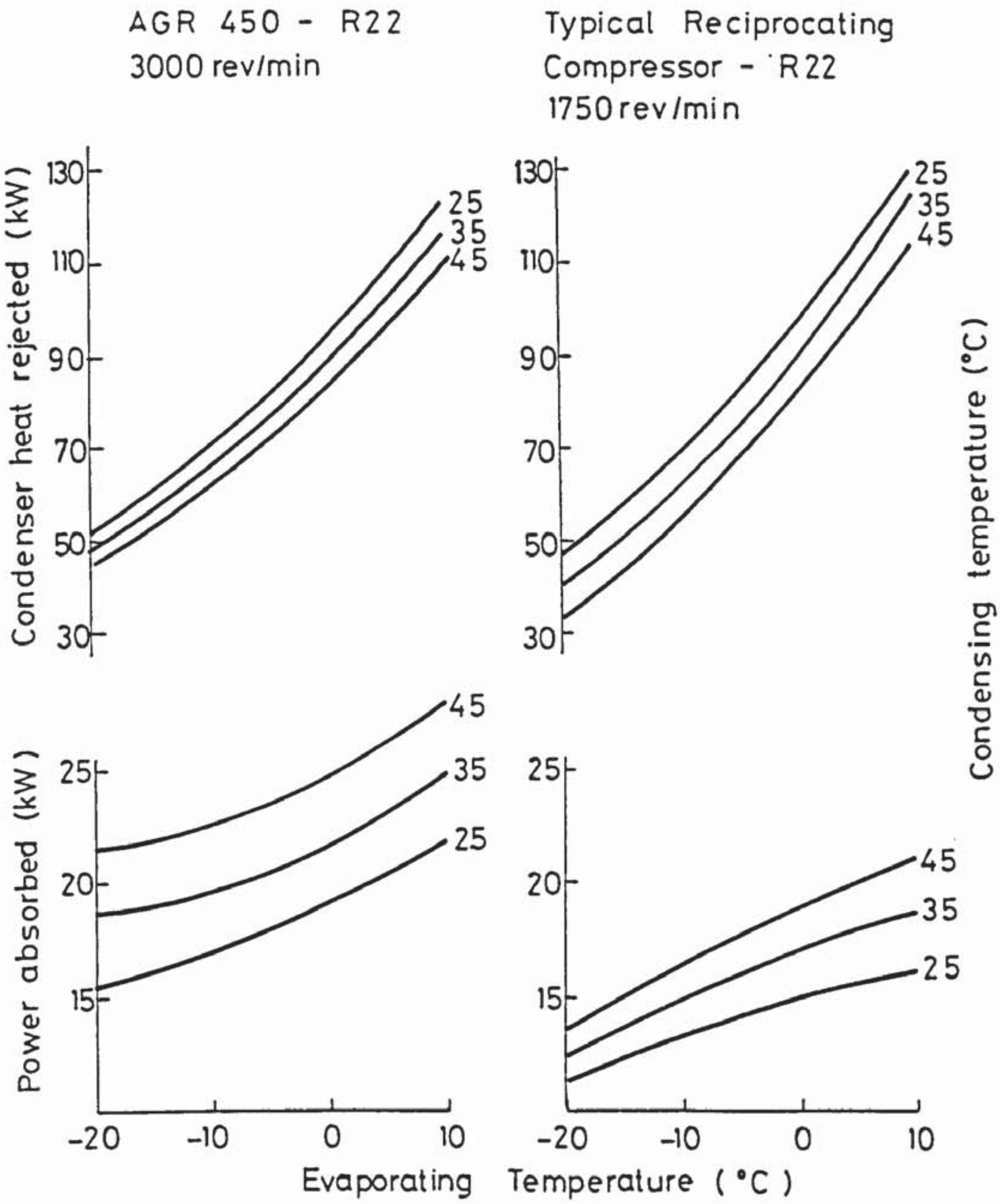


Figure 5.7 **Comparison Of Rotary Vane Compressor Performance With A Typical Reciprocating Compressor**

Since low ambient temperature conditions occur on only a few days per year the heat pump is normally operating at less than its maximum output.

If the speed of the prime mover driving the compressor is infinitely variable, it is possible to reduce the rotary compressor speed to less than one third of its maximum speed. Over this range heat output and power input are proportional to speed.

Similarly, the speed of a reciprocating compressor can also be reduced from its maximum to a minimum speed of approximately 50% maximum speed (1750 rev/min to 900 rev/min generally). Again power input and heat output are porportional to speed.

However, because the reciprocating compressor can only be regulated to 50% maximum speed, alternative methods of off-loading are necessary to reduce heat output. The conventional method is to by-pass cylinders of the compressor by opening a by-pass valve which returns the discharged gas to the compressor suction. Although the heat output is reduced in proportion to the number of cylinders unloaded, the power input is not.

Figure 5.8 shows the reduction in power input compared with heat output for a typical reciprocating compressor, where the evaporating temperature is $5^{o}C$ and the condensing temperature is $45^{o}C$. At all other operating conditions the power input should be multiplied by a

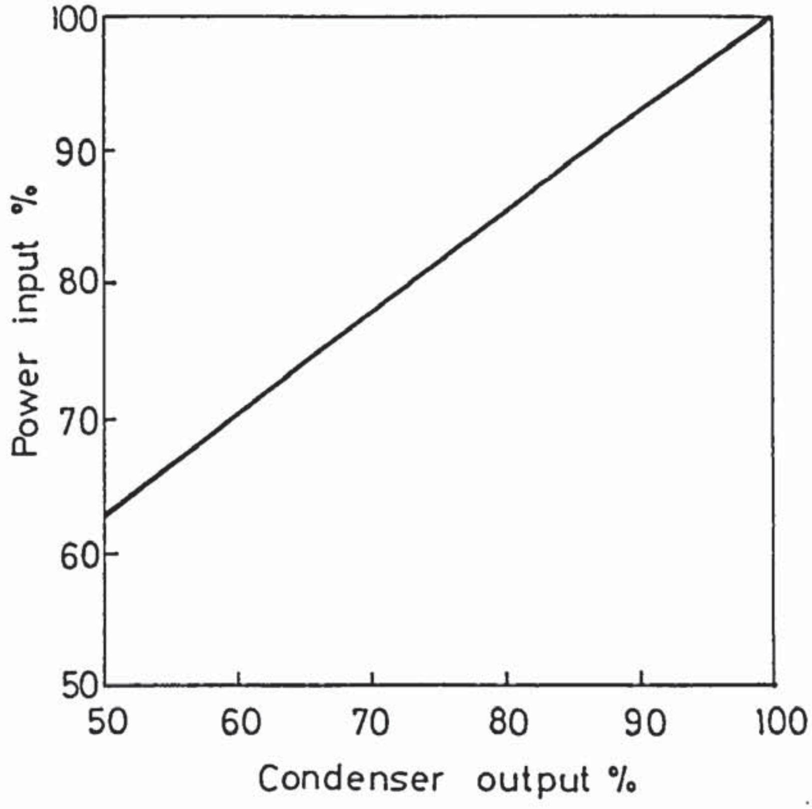


Figure 5.8 Capacity Modulation Of A Reciprocating Compressor Using Cylinder Off Loading [78]

temperature compensating factor F [79] where:

$$F = 0.877\ (1-0.0028T_e)(1+0.0035T_c) \qquad (5.4)$$

T_e: Evaporating Temperature;

T_c: Condensing Temperature

Using equation 5.4 it can be shown (Table 5.1) that as the ambient temperature rises the coefficient of performance of the rotary compressor becomes greater than that of the reciprocating machine, and the seasonal coefficient of performance for both machines would be expected to be comparable.

The building heat load for this comparison was taken as 80 kW at 0°C and 0kW at 20°C. The condensing temperature was fixed at 45°C and a 10K temperature approach for the ambient air onto the evaporator was assumed. The average monthly dry bulb temperature was taken from the CIBS Guide Book [80] for London Heathrow 1957-1966.

Rostell et. al. [81] observed similar trends when varying the system condensing temperature. At low temperature lifts between heat source and heat sink the coefficient of performance for a rotary machine was significantly higher than that for a reciprocating machine. This was achieved without cylinder unloading.

Month	Average Dry Bulb Temp. (°C)	Building Heat Load kW	Evaporating Temp. (°C)	C.O.P.	
				Rotary Compressor	Reciprocating Compressor
Sep	15.2	14.4	5.2	3.73	2.62
Oct	11.9	24.3	1.9	3.53	3.38
Nov	7.4	37.8	-2.6	3.29	3.39
Dec	5.5	43.5	-4.3	3.18	3.59
Jan	3.7	48.9	-6.3	3.01	3.40
Feb	5.1	44.7	-4.9	3.11	3.55
Mar	6.8	39.6	-3.2	3.22	3.69
Apr	9.6	31.2	-0.4	3.41	3.22
May	12.9	21.3	2.9	3.60	2.45

Table 5.1 **Compression Of System C.O.P. For A Rotary And Reciprocating Compressor During A Typical Heating Season**

Seasonal coefficient of performance:

$$\overline{COP} = \frac{\Sigma\, COP}{n} \qquad (5.5)$$

Where n is the number of months in the heating season.

a) Rotary Compressor:

$$\overline{COP} = 3.34$$

b) Reciprocating Compressor:

$$\overline{COP} = 3.25$$

5.7 SUMMARY

The discrepancy between the measured power absorbed, and the power absorbed calculated from the refrigerant properties and thermodynamic considerations, is shown to be due to an increase in refrigerant mass flow rate at compressor discharge compared with compressor suction. This is because liquid refrigerant is injected into the compression chamber midway through the compression process for cooling purposes.

Since the compressor power absorbed is a function of both mass flow rate and specific enthalpy, this increased mass flow rate results in a reduced specific enthalpy at compressor discharge, suggesting an apparent reduction in entropy during the compression process.

By suppressing this liquid injection the measured power absorbed and the calculated power absorbed, from the refrigerant properties, were comparable. The isentropic efficiency of the machine was of the order of 70%.

It is shown that with a 16% to 23% increase in mass flow rate at compressor discharge, using liquid refrigerant injection, the derived power is comparable with the measured power. This is equivalent to only a 0.24% increase in volumetric flow rate.

The only penalty liquid injection introduces into system performance is the small amount of latent heat not absorbed by the interstage liquid circulating between the condenser and compressor.

When compared with a typical reciprocating compressor, the coefficient of performance of the reciprocating machine is higher than that of the rotary compressor, at full load conditions. However, because of the wide range of ambient temperatures experienced during a heating season, and the necessity to off-load cylinders of a reciprocating compressor, the seasonal coefficient of performance of the rotary compressor would be expected to be comparable with the seasonal coefficient of performance of the reciprocating machine.

CHAPTER SIX

EVALUATION OF ENGINE HEAT RECOVERY EQUIPMENT

6.1 INTRODUCTION

A Bowman heat recovery system was used for the production gas engine heat pumps because of problems associated with the reliability and efficiency of the Serck heat recovery equipment. Initial performance checks on this Bowman heat recovery equipment, indicated that the engine heat recovery was still considerably less than predicted by Bissel and Read [64], and a typical temperature distribution around this heat recovery equipment is shown in Figure 6.1.

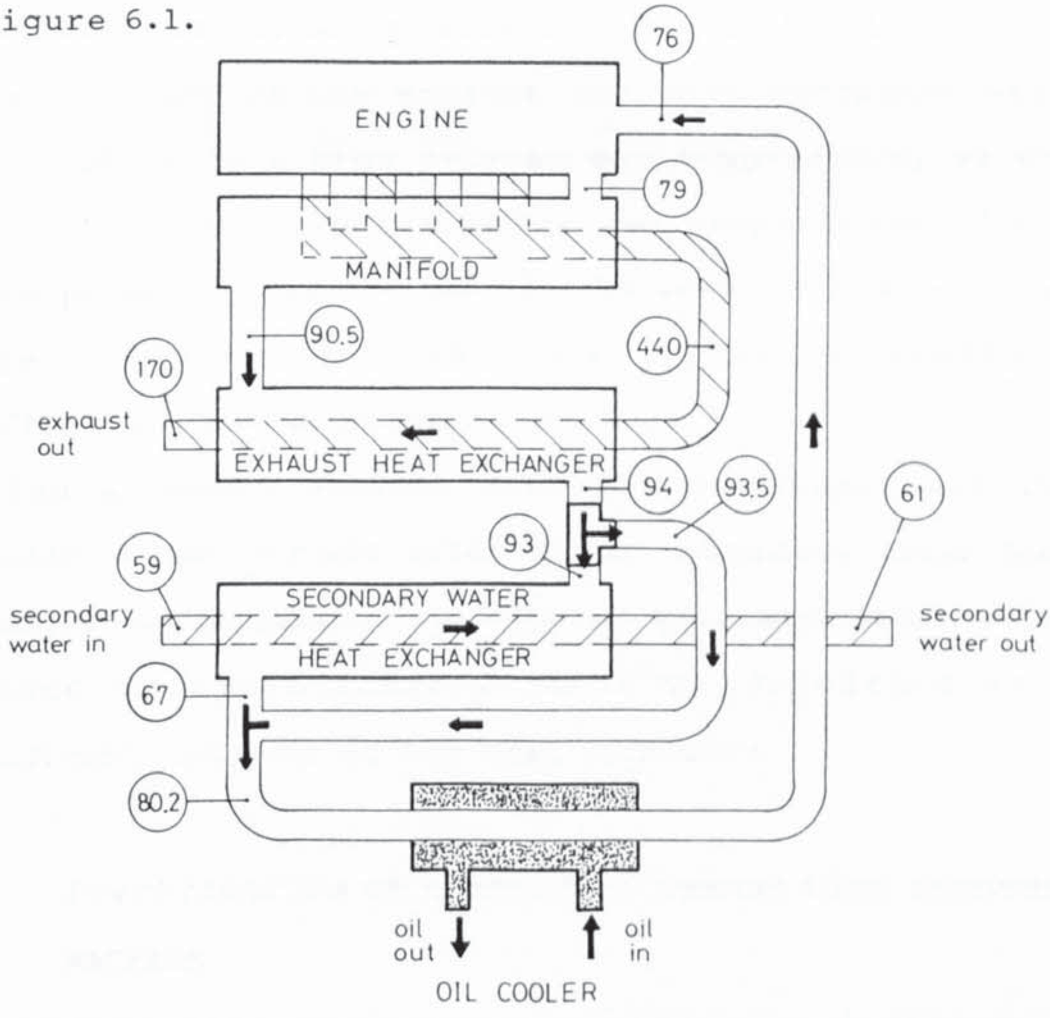


Figure 6.1 Typical Temperature Distribution Around A Standard Bowman Engine Heat Recovery Package. (Engine Speed 2700 Rev/Min: Ambient Temperature 11.3°C) (All Temperatures In °C)

The primary water temperature rise of 3K through the engine was lower than the 15K indicated by Bissel and Read [64]. This was due to additional convection caused by the high air flow rate across the engine resulting from the location of the evaporator in the system, and to radiation.

In addition, the exhaust gas was found to be rejected to waste at a temperature of 170-175°C. This is high compared with the manufacturer's published information, which is based on cooling the exhaust gas to 15°C (Section 4.2.2).

It is, however, apparent from Figure 6.1 that increasing the efficiency of the exhaust gas heat exchanger will still result in a high exhaust gas temperature, as the exhaust cannot be cooled below the temperature of the engine primary water (90-100°C). To lower the exhaust gas temperature below this point would necessitate modifications to the system.

Fitting a second exhaust gas heat exchanger into the secondary water circuit prior to the secondary water heat exchanger would enable the exhaust gas temperature to be lowered to approximately 60-70°C, resulting in a significant increase in the heat recovered.

6.2 INVESTIGATION OF A MODIFIED BOWMAN HEAT RECOVERY PACKAGE

The Bowman heat recovery equipment was modified to incorporate a second exhaust gas heat exchanger, and a typical temperature distribution around this modified equipment is shown in Figure 6.2.

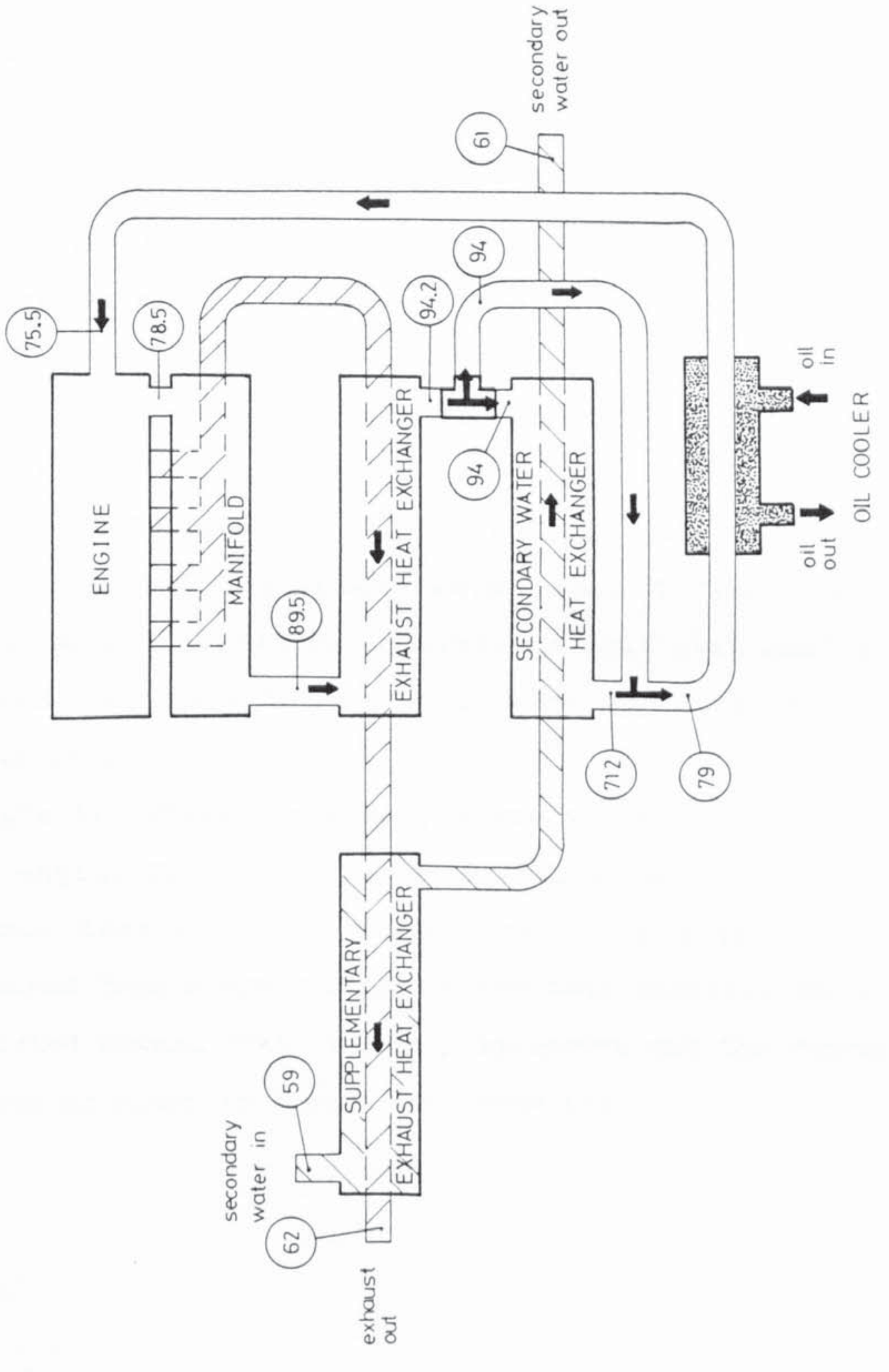


Figure 6.2 Typical Temperature Distribution Around The Modified Bowman Heat Recovery Package (Engine Speed 2700 Rev/Min: Ambient Temperature $9^{o}C$) (All Temperatures In ^{o}C)

A flow meter was fitted into the secondary water circuit and the heat recovery was measured by means of the cooling water flow rate, and differential temperature (between inlet to the exhaust gas heat exchanger and the outlet from the secondary water heat exchanger) for the full engine speed range at various engine load factors.

Comparison of Figure 6.2 with Figure 6.1 indicates that the temperature of the exhaust gas has been considerably reduced from 170°C to 62°C. However, the temperature rise of the coolant passing through the engine is still lower than anticipated. Repositioning the evaporator remote from the engine-compressor module would alleviate this problem. In addition, if the natural gas pipework, and the electronic equipment were removed from the main enclosure, it would be possible to seal this enclosure, thereby retaining the convected heat and improving plant efficiency.

Figure 6.3 shows the steady state system output for the gas engine driven heat pump incorporating the modified Bowman heat recovery equipment. This graph has been obtained from a combination of the heat recovery using the modified Bowman heat recovery equipment and the condenser output as shown in Figure 4.1 (page 63).

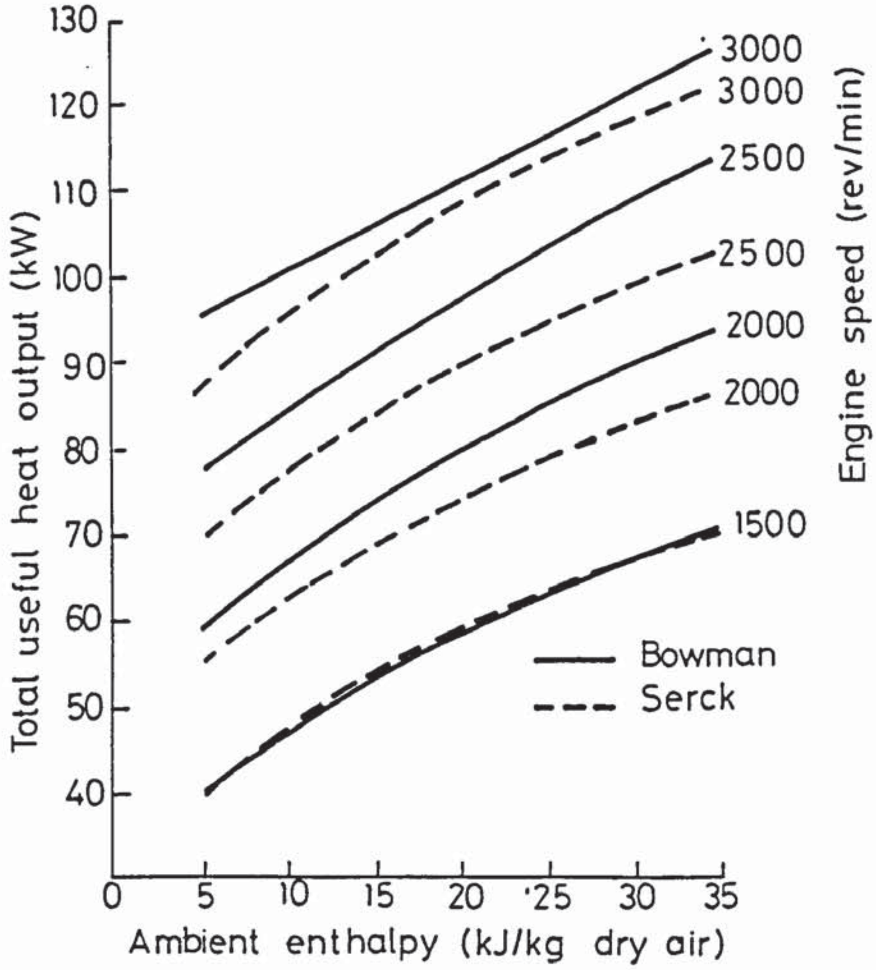


Figure 6.3 **Steady State Total Useful Heat Output v Ambient Enthalpy. Comparison Of The Modified Bowman Equipment With The Serck Heat Recovery Equipment.**

The performance of the gas engine driven heat pump utilising the Serck heat recovery equipment is superimposed onto Figure 6.3 highlighting the better performance of the improved Bowman equipment.

A heat balance for the engine fitted with the modified Bowman heat recovery equipment (Figure 6.4) indicates that 15-20% of the fuel input to the engine is still rejected to waste.

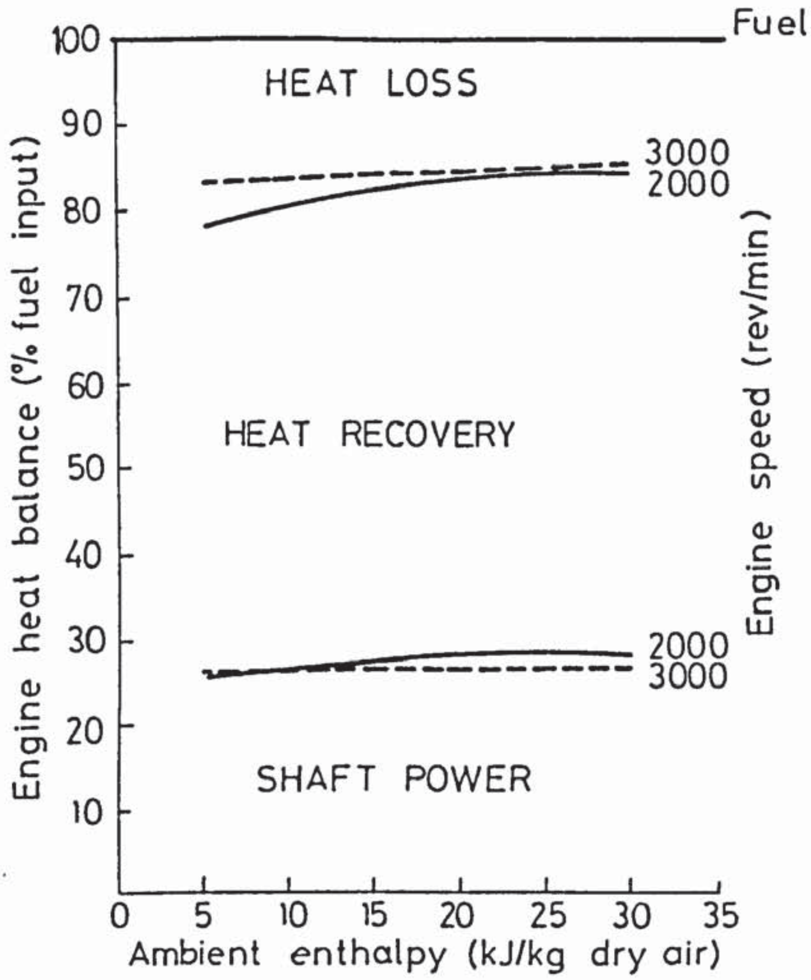


Figure 6.4 **Steady State Heat Balance For The Ford 2274E Engine Fitted With Modified Bowman Heat Recovery Equipment**

Rose and Cooper [82] show that the stoichiometric air/fuel ratio for natural gas is 9.751. Bissel and Read [64] suggest that this is the optimum for the Ford Kent engine and that with a carburettor pressure of 2" W.G., the air/fuel ratio stays near to stoichiometric over the whole speed range. The latent heat content of the exhaust gases, at 15^{o}C, is 9.5% of the heat content of the fuel supplied to the engine under these conditions [82]. In practice it is not feasible to cool the exhaust gases to this temperature.

The economics of recovering some of this exhaust gas latent heat are considered in Chapter 8. Heat at a temperature lower than 60°C would be of use for space heating purposes only, using forced air convection. Condensation of the products of combustion would be prevalent, resulting in a higher corrosion rate of the

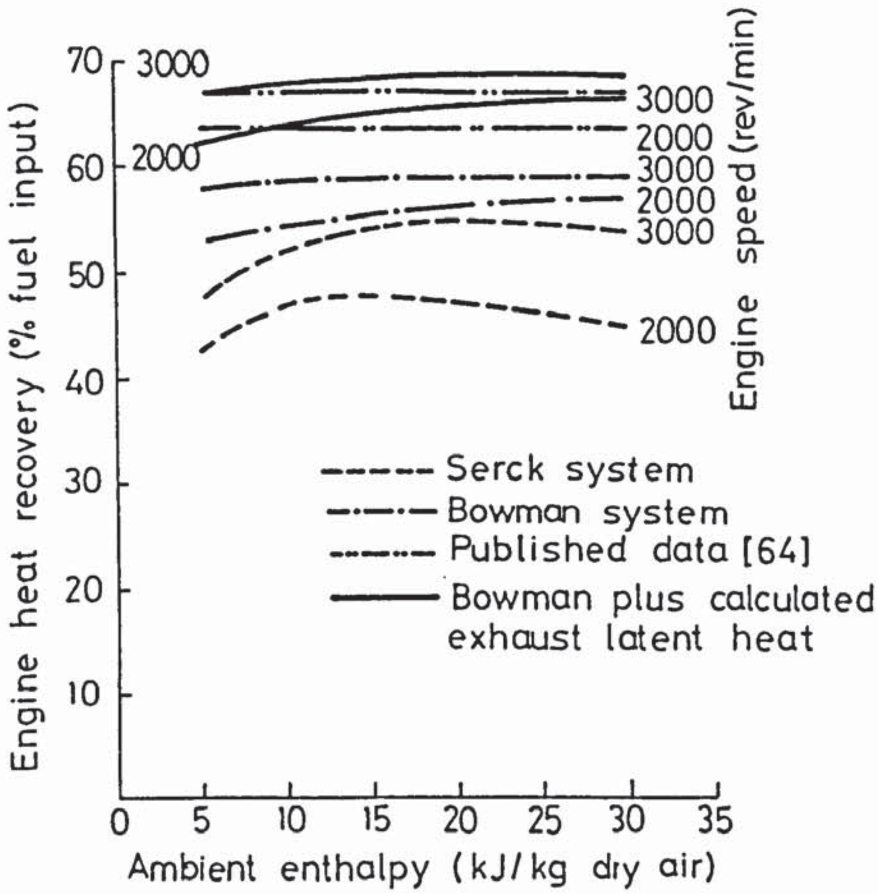


Figure 6.5 **Comparison Of Engine Heat Recovery For The Modified Bowman And The Serck Heat Recovery Systems**

heat exchangers. Since complete combustion of the fuel cannot be guaranteed over a wide range of speed and engine load conditions, air for space heating could not be passed directly over the heat exchanger, as failure of this heat exchanger may result in toxic gases being passed into the heated space, the results of which might be catastrophic. A run-around coil system would have to be utilised, considerably increasing the cost of the plant.

Figure 6.5 compares the heat recovery for the Ford Kent engine fitted with the modified Bowman and the Serck heat recovery equipment. The full load heat recovery as measured by Bissel and Read [64] is superimposed onto this graph. The sum of the heat recovery from the modified Bowman equipment plus the calculated latent heat content of the exhaust is also shown. Close agreement between this and the values quoted by Bissel and Read [64] are obtained which indicates a high efficiency for the modified Bowman heat recovery equipment.

6.3 SUMMARY

Preliminary investigations of a standard Bowman heat recovery package indicated that:

a) The exhaust gases were insufficiently cooled, resulting in a loss of useful engine heat.

b) Convection and radiation from the power unit is excessive due to high air flow rates across the module resulting from the location of the evaporator.

Introducing a second exhaust gas heat exchanger into the secondary water circuit facilitates cooling of the exhaust gas to approximately 60°C, and gives an overall improvement of approximately 7-10% engine heat recovery.

It is essential to position the evaporator remote from the engine/compressor module to eliminate convected heat losses.

Sealing the engine/compressor enclosure would considerably reduce natural convection losses. However both the gas train, and any electronic equipment would have to be located outside this sealed unit. The alternative would be to lag the engine, but this is considered unsatisfactory due to the problems it would create during servicing.

Recovering the exhaust gas latent heat is possible for some applications, using a run-around coil system and cooling to 15°C would result in a further 9.5% of the heat content of the fuel being recovered. The economics of this are considered in Chapter 8.

CHAPTER SEVEN

FROST FORMATION ON AND REMOVAL FROM AIR HEATED EVAPORATORS

7.1 INTRODUCTION

Initial testing of the prototype gas engine heat pump indicated that frost formation on the finned coil evaporator was a major problem. This was because the size of the evaporator fitted to the gas engine heat pump was based on conventional refrigeration and air conditioning practice, using the McQuay data [65]. Carrington [38] claims that this is unsuitable for heat pump applications and that the optimum conditions can only be achieved by increasing the size of the heat exchanger and the volumetric air flow rate.

7.2 CONVENTIONAL METHODS OF FROST REMOVAL

There are numerous conventional methods for frost removal, and the most common are outlined below.

7.2.1 "Reverse Cycle" Operation

This method was developed in the U.S.A. because of the requirement for summer cooling and winter heating. By using a reversing valve one heat pump unit can perform both operations. The valve operation is such that the evaporator becomes the condenser and vice versa.

7.2.2 Electric Resistance Heating

Electric heating elements are laid along the evaporator fins, or inside the tubes. This method is expensive, and results in reduced fin efficiency during normal operation if the elements are attached to the fins.

7.2.3 Warm Water Or Brine Sprays

This is the method commonly used to defrost refrigeration cold rooms, however for heat pump applications the spray would need to be well directed to be effective.

7.2.4 Exhaust Heat

The periodic diversion of the exhaust heat from an internal combustion engine over the evaporator.

7.2.5 Hot Gas Bypass

This is the method most commonly used in Great Britain and Europe. Hot gas from the compressor discharge is directed into the evaporator by means of a magnetic valve, which bypasses the condenser and the expansion valve.
McMullen et. al. [4] have suggested that this has limited defrosting capacity, since once the initial pressure is dissipated, the compressor merely acts as a pump working against minimal pressure, and liquid will inevitably flood back to the compressor.

All of these conventional solutions consider methods to remove frost after its formation, with a subsequent expenditure of energy. As far as the author is aware, there are no recorded methods to prevent the initial formation of this frost, but Delaporte [83,84] has patented an idea which does not use high grade energy for defrosting. Two evaporators are employed, when frost has formed on the first, the refrigerant is diverted to the second, whilst the first defrosts naturally, and vice versa. This method is however only effective when the ambient temperature is above freezing point.

7.3 PROPOSED FLUIDISED BED SOLUTION FOR FROST REMOVAL

The initial formation of frost on an air heated evaporator is advantageous, since not only is the enthalpy of condensation extracted from the water vapour, but also the enthalpy of freezing. It is the subsequent build up of the frost layer which is detrimental to heat transfer.

In order to take advantage of the potential latent heat contribution, a gas fluidised bed system is proposed for use with air heated evaporators. There are two main advantages:

a) The particle circulation inherent in a gas fluidised system produces a mechanical scraping effect on the heat exchanger surface, thereby allowing initial frost formation, but eliminating the build up of a frost layer.

b) The heat transfer coefficient is increased (previous work by the author [85] using an electrically heated horizontal tube indicates at least a six fold increase) resulting in increased system performance.

7.4 NOTES ON FLUIDISATION

A fluidised bed is a system containing small solid particles, generally sand, or metal powder, of less than 1 mm diameter. If a fluid is forced upwards through a bed of such particles, the particles offer resistance to the fluid flow. As the flow rate increases the bed will expand until a point is reached where the drag forces exerted on the particles, are sufficient to support the weight of the particles, and in this state the system will behave as a fluid. The pressure drop through the bed will be equal to the weight per unit area of the particles in the bed.

If the fluid is a gas, and its flow rate is further increased, the system becomes unstable and cavities containing few particles are formed, which look like bubbles rising through the bed. These bubbles are responsible for inducing particle circulation within the bed, and it is this circulation which enhances the heat transfer properties of a gas fluidised system. Beyond a certain flow rate, the drag forces are such that the particles become entrained within the fluid stream and are carried from the bed.

Smaller particles tend to become entrained at lower fluidising velocities, and particle density is also an influencing factor.

Figure 7.1 shows the possible advantages and limitations of a fluidised bed system applied to evaporator defrosting.

7.4.1 Inherent Problems Of Fluidised Beds

Four inherent problems exist in gas fluidised beds:

a) Channelling: Botterill [86] suggests that gas fluidised beds, of fine irregular particles are particularly prone to this, due to the relatively strong inter-particle forces existing. As the fluid flow rate through the bed is increased, instead of the bed expanding uniformly the fluid may tend to open up channels through the bed. Then having created a path of low resistance, the fluid tends to follow this

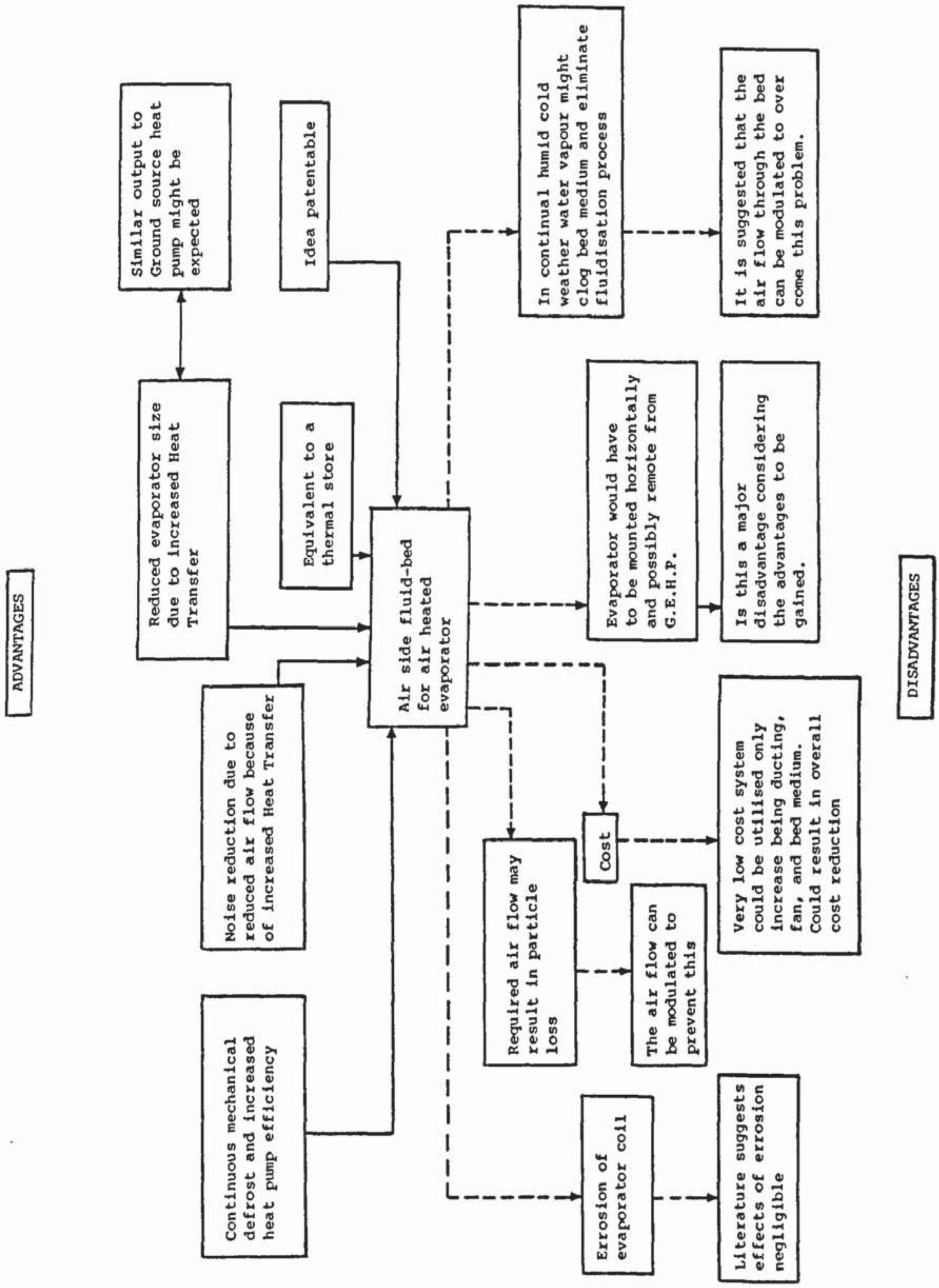


Figure 7.1 **Possible Advantages And Limitations Of A Fluidised Bed Defrosting Mechanism**

path. Further increase in the fluid flow velocity enlarges this path, until in the extreme case, the bed fails to fluidise at all.

b) Slugging: In gas fluidised beds, there is a tendency for bubbles to grow and coalesce as they rise through the bed, until they occupy the whole cross sectional area of the bed as slugs. This is particularly noticeable at high fluid flow velocities in deep narrow beds. The gas "slugs" rise through the bed carrying the particles ahead of them until instability occurs and the solids collapse back into the bed.

c) Segregation: When a bed of particles with a wide range of density, or particle size is fluidised, the large dense particles tend to sink to the bottom, and the smaller less dense particles to float to the top of the bed.

d) Solids Transport: Some bed material is lost, when particles are thrown upwards into the space above the bed as bubbles break the surface.

   Botterill [86] suggests that the highest ejection velocity is when bubbles coalesce just before breaking the surface. Thus if the freeboard above the expanded bed is too small, many particles may have sufficiently high velocities on leaving the surface to carry them out of the duct. Even with a large freeboard, the fines entrained in the gas stream will be carried out of the bed.

7.4.2 Heat Transfer Mechanism

The high degree of mixing generated by a rising bubble in the bed means that the bulk of the bed will be at a uniform temperature. High rates of heat transfer are obtainable between an exposed surface and the gas fluidised bed, so facilitating the addition and removal of heat from the bed as required.

Botterill et. al. [87] suggest that the true character of the process is one in which the dominating factor is the much greater ability of the particles to transfer heat than the gas, by virtue of their very different volumetric heat capacities (of the order of one thousand fold difference). The gas fluidised system therefore represents a fluid of high heat capacity, and low vapour pressure.

Heat transfer within the bed itself is dependent upon three mechanisms:

a) Fluid-Particle Heat Transfer: The average fluid-particle heat transfer coefficients based upon the total particle surface area are often not very large, but a fluidised bed is capable of exchanging heat very effectively when fluidising a gas, because of the very large surface area exposed by the particles. Juveland et. al. [88] suggest that this is in the order of 3000-4500 m^2/m^3.

b) Particle Heat Transfer: This is important during start-up, and accelerates the creation of a uniform bed temperature.

c) Radiated Heat Transfer: Botterill [86] suggests that radiation will not have any noticeable effects until the temperatures are in excess of 600^{o}C.

7.4.3 Heat Transfer To Immersed Surfaces

Previous work by the author [85] indicates that at least a 6 fold increase in heat transfer coefficient can be expected for a horizontal tube in a fluidised bed, compared with forced convention. Botterill [86] suggests that heat transfer coefficients are dependent on the bed material and on the fluidising conditions. Poorer coefficients are to be expected if the particles are large and dense, and the fluidising gas velocity is low.

The effect of finned surfaces in a fluidised bed is to increase heat transfer, from that with a plain tube. Botterill [86] shows that heat transfer will increase with the height of fins up to about 25 mm, and is also sensitive to inter tube spacing, up to about 50 mm between tubes.

7.5 COMMERCIAL FEASIBILITY OF A FLUIDISED BED DEFROSTING MECHANISM FOR A HEAT PUMP EVAPORATOR

Adding a fluidised bed to an evaporator will increase the capital costs of a heat pump due to the additional components required. However, if the conventional defrosting process could be eliminated the energy saved would partially offset these additional costs.

7.5.1 Heat Transfer Characteristics

a) Plain Horizontal Tube: Forced Convection

For a plain horizontal tube it can be shown that:

$$\frac{1}{U} = \frac{1}{h_a} + \frac{1}{h_r} \qquad (7.1)$$

(For further details see Simonson [89])

Where:

U = overall heat transfer coefficient
h_a = air side heat transfer coefficient
h_r = refrigerant side heat transfer coefficient

For the prototype gas engine heat pump the average air velocity across the evaporator was 3.5 m/s.

At this condition h_a = 22.7 W/m^{2}K [90]

and for R22 boiling h_r = 2890 W/m^{2}K [91]

Thus:

U = 22.5 W/m^{2}K

From this it can be seen that the heat transfer coefficient on the air side has the most influence.

b) Finned Horizontal Tube : Forced Convection

The most common method used to improve heat transfer is to fit fins to the outside of the tube thereby increasing the heat transfer surface area.

It can be shown that for a finned horizontal tube:

$$\frac{1}{U} = \frac{1}{h_a(1+\eta A_s)} + \frac{1}{h_r} \qquad (7.2)$$

(referred to the external surface. For further details see Simonson [89])

Where:

A_s = Fin surface area per unit primary surface area

η = Fin efficiency

From Appendix 5.4, A_s is 4.

Assuming a fin efficiency of 70%

$U = 83.7\ W/m^2K$

This represents a 3.7 fold increase in heat transfer coefficient when compared with a plain horizontal tube.

c) Plain Horizontal Tube in a Fluidised Bed

It has been shown that for a plain tube, a 6 fold increase in the air side heat transfer coefficient can be achieved when using a fluidised bed [85].

Thus from equation 7.1:

$U = 130\ W/m^2K$

d) Finned Horizontal Tube in a Fluidised Bed

It is suggested that the average air side heat transfer coefficient will be between 1.5 and 3 times larger, if fins are added to a plain horizontal tube in a fluidised bed [86].

Hence, assuming a 2 fold increase in air side coefficient, and applying equation 7.1:

$U = 249\ W/m^2K$

It can be seen that when compared with forced convection over a finned tube this is equivalent to a 3 fold increase in overall heat transfer coefficient.

7.5.2 Fluidised Bed Design Considerations

a) Minimum Fluidising Velocity:

Correlations by Davidson et. al. [92] from experimental data show that:

$$Re_{mf} = 25.7 \quad \{ \sqrt{(1 + 5.53 \times 10^{-5}\ Ga)} - 1 \} \qquad (7.3)$$

Where:

Re_{mf} = Reynolds no. for incipient fluidisation = $\dfrac{U_{mf}\, d_p \rho_a}{\mu_a}$

Ga = Galileo no. = $\dfrac{\rho_a\ (\rho_p - \rho_a)\ g d_p^{\,3}}{\mu_a^{\,2}}$

ρ_a = air density
ρ_p = particle density
g = acceleration due to gravity
d_p = particle diameter
μ_a = air viscosity
U_{mf} = minimum fluidising velocity

Consider silica sand 0.5 mm diameter with a density of 1650 kg/m^3 [85]. For dry air at 0^oC Mayhew et. al. [93] state:

ρ_a = 1.284 kg/m^3 ; μ_a = 1.725 x 10^{-5} kg/ms

Thus from equation 7.3:

U_{mf} = **0.15 m/s**

In practice, velocities of 10 U_{mf} are used to ensure good fluidisation and to avoid particle carryover.

Hence an air velocity of 1.5 m/s is used in this study.

b) Evaporator Design:

$$Q_E = UA_E\, \Delta T_m = \rho_a \dot{u}_a C_p \Delta T_a \qquad (7.4)$$

Where:

Q_E = Evaporator heat extraction rate
U = Overall heat transfer coefficient
A_E = Evaporator surface area
ΔT_m = Log mean temperature difference
ρ_a = Air density
$\dot{u}_a$ = Air flow rate
C_p = Specific heat capacity
ΔT_a = Differential air temperature

Assuming that:

i) density and specific heat capacity of the air remain constant

ii) temperature differentials are unchanged

iii) surface area A is proportional to face area multiplied by the number of rows

Then for the conventional finned evaporator:

U_C = 83.8 W/m^2K
A_{Ec} = 1.55 x 4 = 6.2 m^2
$\dot{u}_{ac}$ = 3.35 m/s

and for the fluidised bed evaporator:

U_f = 249 W/m^2K
A_{Ef} = to be determined
$\dot{u}_{af}$ = 1.5 m/s

To maintain a sufficient volume of air across the evaporator the face area of the evaporator must be increased for the fluidised bed condition. Thus:

Face Area (A_{EF}) = 1.55 x $\frac{3.35}{1.5}$ = 3.5m^2

However, since the overall heat transfer coefficient is much higher for a fluidised bed, the total heat transfer surface can be reduced - (i.e. by reducing the number of rows).

Then for the fluidised bed:

$$A_{EF} = \frac{83.8 \times 6.2}{249} = 2.1m^2$$

The increased heat transfer in a fluidised bed indicates that a single row evaporator 2.1m^2 would be adequate. However, the requirement to maintain the volumetric air flow rate with a lower face velocity necessitates the use of a single row evaporator 3.5m^2.

c) Distributor Plate:

Harrison et. al. [92] have proposed numerous distributor plate designs, but the simplest and most easily obtained is small gauge mesh. This distributor will have approximately 50% open area.

Assuming that each hole in the distributor will act as a simple orifice, the sum of the areas of each orifice will be equivalent to one orifice of that total area.

To determine the distributor pressure drop from BS 1042 Part A [94]:

$$N_a = \frac{\dot{m}_a}{0.01252\ Do^2 \sqrt{h_d\ \rho_a}} \tag{7.5}$$

$$mE = 1.65\ N_a \tag{7.6}$$

and

$$\frac{do}{Do} = [(mE)^2/\{1 + (mE)^2\}]^{1/4} \tag{7.7}$$

(See nomenclature for definition of symbols)

Thus, for a mesh $3.5m^2$ with 50% open area:

$mE = 0.258$; $N_a = 0.156$

$h_d = 14.8$ mm H_2O

The pressure drop through a fluidised bed is equal to the weight per unit area of particles in the bed [86]. Hilby [95] and Zuiderweg [96] indicate that the distributor pressure drop should be 10-15% of the bed pressure drop. For a fluidised bed of 0.5 mm diameter silica sand particles 75 mm deep this is equivalent to 7.5-11.5 mm H_2O. Although the pressure drop for a mesh distributor slightly exceeds this value, it should not present any problems.

7.5.3 Cost Analysis

Various cost savings are available from the original system, as well as energy savings by the elimination of the defrosting process. These are outlined below:

a) Cost Savings From Forced Convection Evaporator With Hot Gas Bypass for Defrosting

Evaporator 1.55 m^2, 4 rows, 8 fin/inch:	660.00
Propellor fans, 2 @ £90.00	180.00
Hot Gas Bypass	100.00
TOTAL	£940.00

b) Energy Savings

Propellor fans, 2 @ 0.75 kW	1.5 kW
Frosting and Defrosting	15 kW
TOTAL	16.5 kW

c) Increased Cost Of Fluidised Bed Equipment

Evaporator 3.5 m^2, single row, 8 fin/inch	375.00
Silica Sand 260 kg, 0.3-0.6 mm dia.	30.00
Distributor plate 3.5 m^2 M.S. 40:34	50.00
Centrifugal fan 11000 cfm, 90 mm H_2O pressure drop	900.00
Ductwork 3.5 m^2 C.S.A. x 2 m high	100.00
TOTAL	£1455.00

d) Increased Energy Usage

Centrifugal Fan 6 kW

e) Fluidised Bed Payback

Based on 3000 annual operating hours and a gas cost of 1.04 pence/kWh then the annual energy saving is:

$$\text{annual energy cost saving} = (\text{energy saving} - \text{energy usage}) \times \text{annual operating hours} \times \text{fuel cost}$$

$$= (16.5 - 6) \times 3000 \times 1.04 = £327.60$$

and

$$\text{Payback} = \frac{\text{increase in capital cost}}{\text{annual energy cost savings}}$$

$$\textbf{Payback} = \frac{1455 - 940}{327.6} \quad < 2 \textbf{ years}$$

7.5.4 Conclusions Drawn From The Feasibility Study

For the prototype gas engine driven heat pump the heat lost due to frost formation and its subsequent removal is in the order of 10 to 15 kW at the rated capacity (using the data quoted by Heap [6] for a similar system).

The capital cost of the gas engine driven heat pump would be increased using a fluidised bed, but the energy saved by the elimination of the hot gas defrosting process would result in a payback period of less than two years. From this it is apparent that a fluidised bed evaporator is commercially viable providing that mechanical scraping by the fluidised solids eliminates the necessity for supplementary defrosting.

7.6 PROTOTYPE GAS FLUIDISED BED SYSTEM FOR EVAPORATOR DEFROSTING

Due to the high cost and inconvenience involved in fitting a fluidised bed to the existing gas engine heat pump, a prototype test rig incorporating an electric heat pump was designed. The complexity of the test rig was kept to a minimum as the only reason for the prototype was to test the theory of mechanical defrosting. It was envisaged that any major changes necessary could be made when designing a fluidised bed to suit the gas engine heat pump system.

Figure 7.2 is a schematic diagram of the prototype fluidised bed, which could be operated either in the fluidised bed mode or simply with forced air convection.

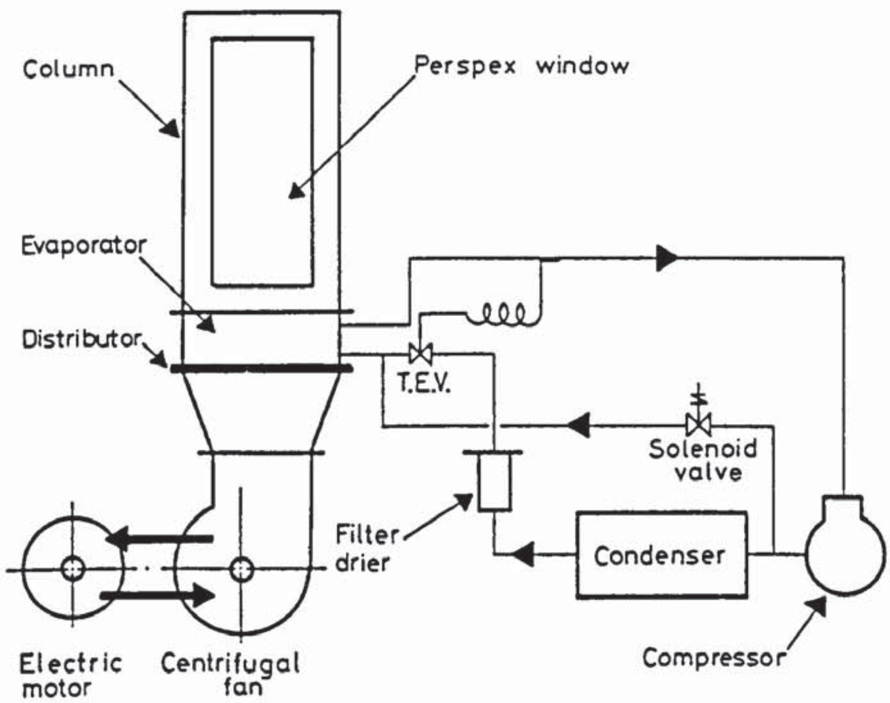


Figure 7.2 Schematic Diagram Of The Prototype Fluidised Bed

This prototype can be divided into four main parts:

a) The Electric Heat Pump:

This was a vapour compression machine. The compressor was a single cylinder hermetic unit suitable for refrigerant R12. The evaporator was a two row finned coil which was mounted horizontally in the fluidised bed test section.

The condenser was water cooled, and the liquid refrigerant was metered to the evaporator by means of a thermostatic expansion valve. The refrigerant pipework was sized in accordance with the procedure outlined by Trane [97].

A hot gas bypass line, controlled by a magnetic solenoid valve, was incorporated into the system to facilitate defrosting when operating in the forced convection mode.

b) Fluidised Bed Test Section:

The plenum chamber was made from mild steel plate of welded construction, designed to suit the cross sectional area of the fan delivery, and the face area of the evaporator. The fluidising duct section was made from mild steel plate of welded construction with a perspex insert, which enabled the fluidisation process to be observed. Stainless steel mesh supported by a 16 SWG perforated plate was used for the distributor.

c) The Fluid Circulating Section:

The fan was of the centrifugal type, necessary to overcome the flow resistance of a fluidised bed system. This fan was belt driven by a three phase squirrel cage induction motor. A belt drive was selected because of the necessity to operate the fan over a range of speeds, and changing pulley sizes was considered to be the most economical method of speed control.

d) Ancillary Measuring Equipment:

There are 5 main variables to be measured in this system.

i) Temperature: Proprietary thermocouples, linked to an "Edale" multi-position temperature measuring device, were used to measure temperature variations around the system.

ii) Pressure: Bourdon type pressure gauges were used to measure the pressure variation around the heat pump system.

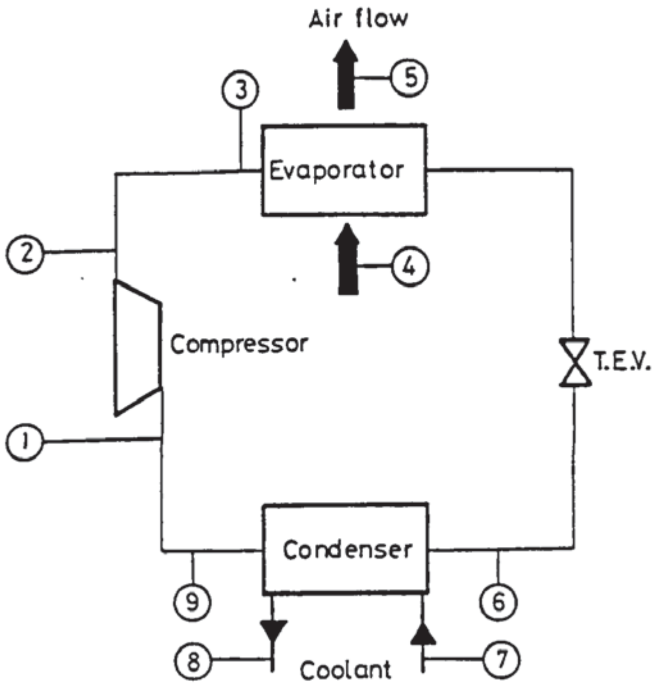
iii) Electrical Power: A portable "Compton" wattmeter was used to measure compressor power absorbed.

iv) Flow: The volumetric flow rate of the fluidising medium was measured using an "Airflow Developments" anemometer.

The condenser cooling water volumetric flow rate was measured using a turbine meter of the type discussed in Chapter 3.

v) Humidity: The wet and dry bulb temperature of the air onto the evaporator was measured using a sling hygrometer.

Figure 7.3 is a schematic diagram showing the location of this instrumentation.



Instrumentation

1. Refrig Temp and Pres
2. Refrig Temp and Pres
3. Refrig Temp and Pres
4. Air Temp and Humidity
5. Air Temp and Flow
6. Refrig Temp and Pres
7. Cool Water Temp
8. Cool Water Temp & Flow
9. Refrig Temp and Pres

Figure 7.3 Instrument Layout : Schematic

7.7 FLUIDISED BED TEST PROGRAMME

The objective of this test programme was to evaluate the fluidised bed defrosting mechanism, so the reactor was operated in both the forced convection mode, with the bed particles removed, and in the fluidised state.

7.7.1 Instrumentation Calibration

The Bourdon tube gauges were calibrated against N.P.L. calibrated Bourdon tube test gauges, using pressurised nitrogen, and the thermocouples were calibrated using the technique outlined in Section 3.7.1. Results of these calibrations are contained in Appendix 3. All other instrumentation was calibrated by the manufacturers.

7.7.2 Experimental Procedure : Forced Convection Mode

The pulley arrangement between the electric motor and the centrifugal fan was set to give the design air flow rate. This air flow rate was measured using a proprietary anemometer.

The condenser cooling water flow rate was regulated by means of a shut off valve to give a condensing pressure of approximately 12 bar absolute. The system was allowed to run until steady state conditions were reached. A defrosting process was initiated to clear any frost formation from the evaporator. On termination of the defrosting process the values of the parameters shown in Figure 7.3 were recorded. The tests were repeated at five minute intervals for approximately 90 minutes. A condensing pressure of approximately 12 bar absolute was maintained for the duration of the experiment.

7.7.3 Experimental Procedure : Fluidised Bed Mode

The fluidising medium was added to the reactor, and the pulley arrangement between the electric motor and the centrifugal fan was set to give the design air flow rate, which was checked using the proprietary air flow meter. The foregoing procedure was repeated but without allowing steady state to be reached.

7.7.4 Presentation and Discussion Of Results

Silica sand with a particle size range of 300-600 μm was selected as the fluidising medium. The air flow was checked and adjusted to the design condition. Very poor air distribution was experienced, with approximately 30% of the bed vigorously agitated, and the remainder only lightly.

The system ran for approximately 15 minutes, but during this time the bed progressively settled until there was no measurable air flow through the bed. On inspection this was found to be due to agglomeration of the sand particles resulting from dehumidification of the fluidising air. From this it was concluded that silica sand was unsuitable as the fluidised bed medium.

A survey of alternative bed materials indicated two possible alternatives:

i) P.T.F.E. of mean particle size 1.5 mm

ii) Glass spheres 2-3 mm diameter

P.T.F.E. particles were considered more suitable since they are hydrophobic. In addition the particle size was more suitable for the fluidised bed test rig which had a fin spacing of 3.2 mm (8 fins per inch).

Since it had been difficult to remove the silica sand from the test section, it was decided to perform the forced convection tests before evaluating the P.T.F.E. fluidised bed. The results of the forced convection tests are shown in Figure 7.4. It can be seen that trends similar to those experienced during the transient testing of the prototype gas engine driven heat pump have been obtained (see Section 4.4). However, the fall off in performance with time due to frost build up is much less pronounced. the most likely reason for this is that the gas engine driven heat pump utilises a low pressure axial fan, compared with the high pressure centrifugal fan fitted to the fluidised bed reactor. The axial fan would be more susceptible to increased flow resistance due to frost formation.

Before the P.T.F.E. was supplied to the reactor, steps were taken to improve the air distribution. A horizontal mesh was fitted across a section of the plenum chamber to increase the pressure drop through that section. The P.T.F.E. was then added, and the air flow rate checked. However, the quality of fluidisation was still poor, again only 30% of the bed was vigorously agitated.

The results of the fluidised bed evaluation using P.T.F.E. are shown superimposed onto the forced convection curves in Figure 7.4. It can be seen that the performance is increased by 20 to 30% compared with the forced convection experiments. However a pronounced fall off in performance is apparent after 70 minutes operation. It was observed

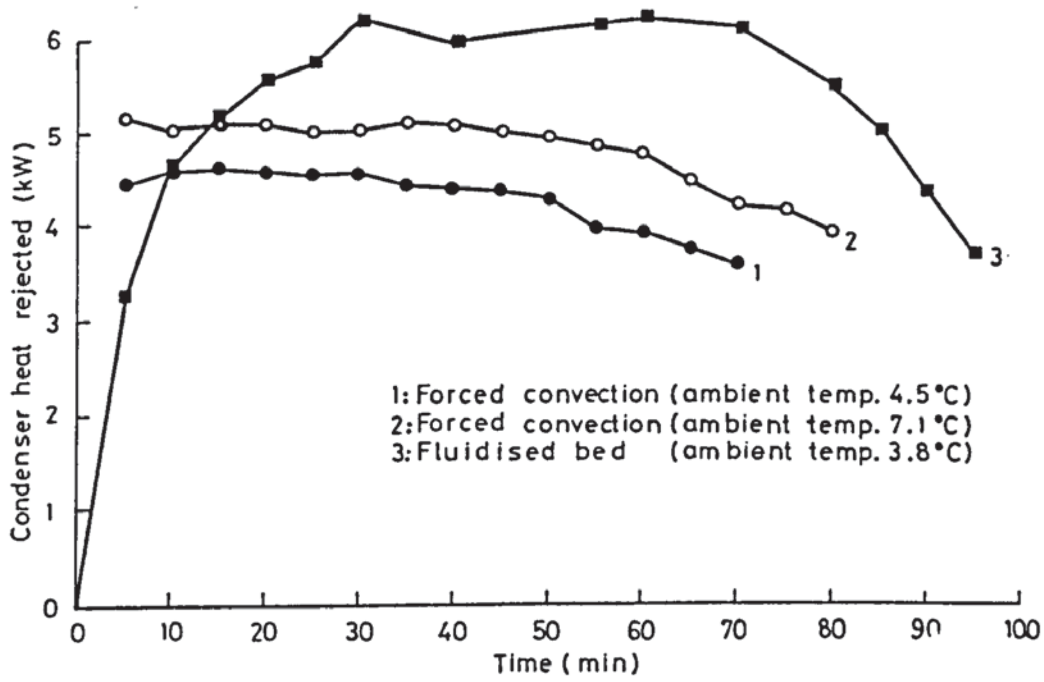


Figure 7.4 Comparison Of The Performance Of An Electric Heat Pump, And A Similar Unit Fitted With A Fluidised Bed Reactor

during testing that fluidisation became progressively more vigorous in one third of the bed, whilst the remainder of the bed settled. Eventually a channel containing very few particles was created in this region, such that fluidisation was no longer prevalent. Effectively one third of the evaporator was subjected to forced air convection. The evaporating pressure fell rapidly, and frost formation on this section of the evaporator was inevitable, resulting in the subsequent loss of performance.

This problem can be attributed entirely to the design of the apparatus.

The increase in performance of 20-30% compared with forced air convection is considerably lower than suggested by the design calculations. However, it should be noted that the ambient temperature during the fluidised bed experiments was lower than during the forced convection experiments. In addition the poor quality of fluidisation considerably reduced the effective evaporator surface.

The air temperature approach to the evaporator was reduced in the fluidised mode, indicating the improved heat transfer characteristics of the system.

During initial testing using the silica sand, some particles adhered to the evaporator surface, and were not removed when the evaporator was cleaned using compressed air. However, when the reactor was switched off at the completion of testing in the fluidised bed mode, these sand particles were observed to be mixed with the P.T.F.E., indicating that the mechanical scraping effect had been successful. In addition a sample of P.T.F.E.

particles taken from the bed showed no signs agglomeration.

Sufficient information has been obtained to suggest that a fluidised bed mechanism is a practical solution to the defrosting problem. However, further development work is now necessary to perfect the technique of fluidisation.

7.8 SUMMARY

A fluidised bed defrosting mechanism has been proposed, the advantages of which are:

i) Improved heat transfer characteristics for an air heated evaporator.

ii) Automatic defrosting due to a mechanical scraping effect produced by particle circulation.

This system is superior to conventional defrosting mechanisms in that there is a net gain rather than loss of energy.

A review of the literature suggested that no similar system had been reported, and a feasibility study indicated that although the fluidised bed reactor resulted in an increased capital cost, the proposed energy savings would result in a payback period of less than two years. Thus a prototype reactor was built to test the hypothesis. The prototype was of a poor design due to a lack of experience of the process. However, sufficient operating data was obtained to suggest that the defrosting mechanism was effective, and that, in addition, a substantial increase in evaporator performance could be obtained.

Further work is however necessary to perfect the technique of fluidisation.

CHAPTER EIGHT

COMMERCIAL CONSIDERATIONS

8.1 INTRODUCTION

In a commercial enterprise it is important to define the market, before costly experimental work is undertaken. A confidential market analysis for the gas engine heat pump was conducted by Denco Air Limited before this project was initiated. The results indicated an increasing demand for heat pumps up to and including 1990.

This chapter considers the essential technical support necessary for this market share to be attained.

8.2 FACTORS AFFECTING THE DESIGN OF THE PRODUCT RANGE

The Denco Air philosophy for manufacture is the production of modular units, and this Company policy was adopted for the development of the gas engine heat pump product range. The engineering division supplies the modular units to numerous contracting divisions, who supervise the installation, including the selection of pipework, ductwork and ancillary equipment. The area of interest in the development of a range of modular units is the engine-compressor module, plus any associated ancillary equipment contained within the engine-compressor enclosure.

The limitations imposed by Denco Air for the modular range were:

a) The prototype Ford Kent/AGR 450 unit should be the smallest module in the range.

b) The largest module should develop a shaft power of no more than 100 kW. For applications requiring higher shaft power, multiple modules would be used.

c) Refrigerants R12 and R22 were stipulated as the working fluids.

8.2.1 Engines For Modular Units

A survey of all the natural gas fuelled engines available in the U.K. was made, and Kompass[98] was used as the source. Besides the foregoing restrictions imposed by Denco Air, the following selection criteria were also initially applied by the author.

a) One manufacturer was required to supply engines for the full range.

b) Engines were required with the ability to operate using multiple fuels.

c) Engines were required to be water cooled.

d) Capital cost.

The fuel requirements were for natural gas L.P.G. and diesel oil. Liquid cooled engines were considered better than air cooled engines for the following reasons:

a) The specific heat capacity of air falls with rising temperature whereas that of a water/anti-freeze solution increases.

b) A liquid coolant allows the engine to retain heat for longer periods than air, so that cylinder wall temperatures do not fall below dew point temperature so quickly, and condensation problems are less likely.

c) The inherent safety margin incorporated in liquid cooling systems is more likely to prevent damage in the event of engine malfunction.

d) It is difficult to service air cooled engines used for stationary applications as accessibility is poor, and dismantling the cooling trunking is necessary even for minor work.

e) Liquid coolant is a positive factor in achieving high durability, because of its better stability and ease of control.

Figures 8.1 and 8.2 are bar charts showing the range of engines supplied by various manufacturers. It should be noted that Power Torque convert Ford engines to operate on natural gas, and are the recognised Ford agents.

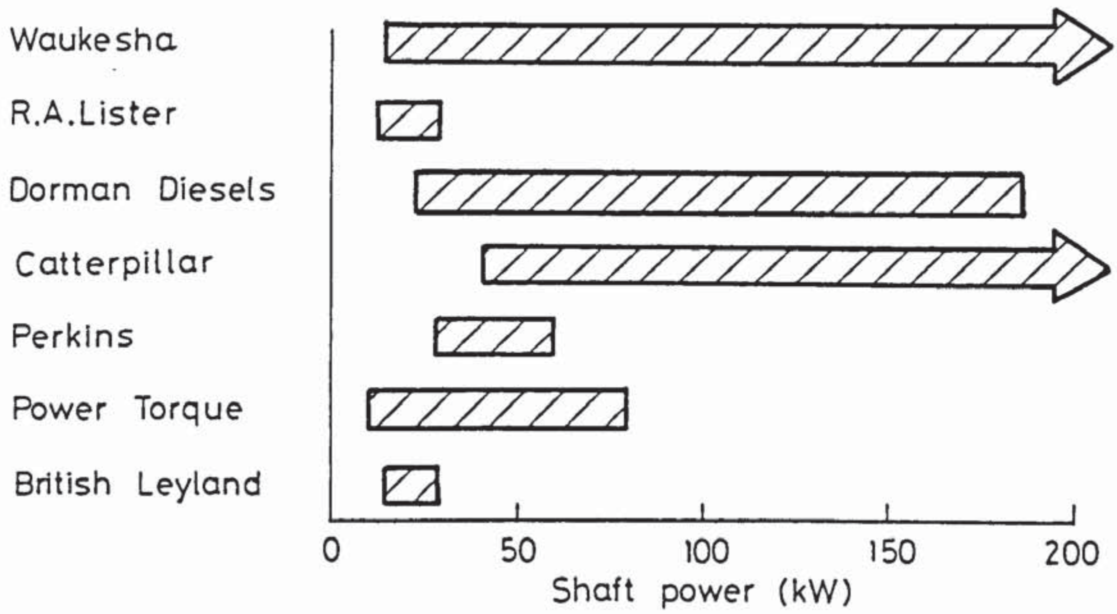


Figure 8.1 Bar Chart Of Natural Gas Engines Currently Available In The U.K.

From these charts the following comments relating to the final selection criteria apply:

a) The following manufacturers were able to supply engines for the full range required:

Ford/Power Torque	Catterpillar
Dorman Diesels	Waukesha

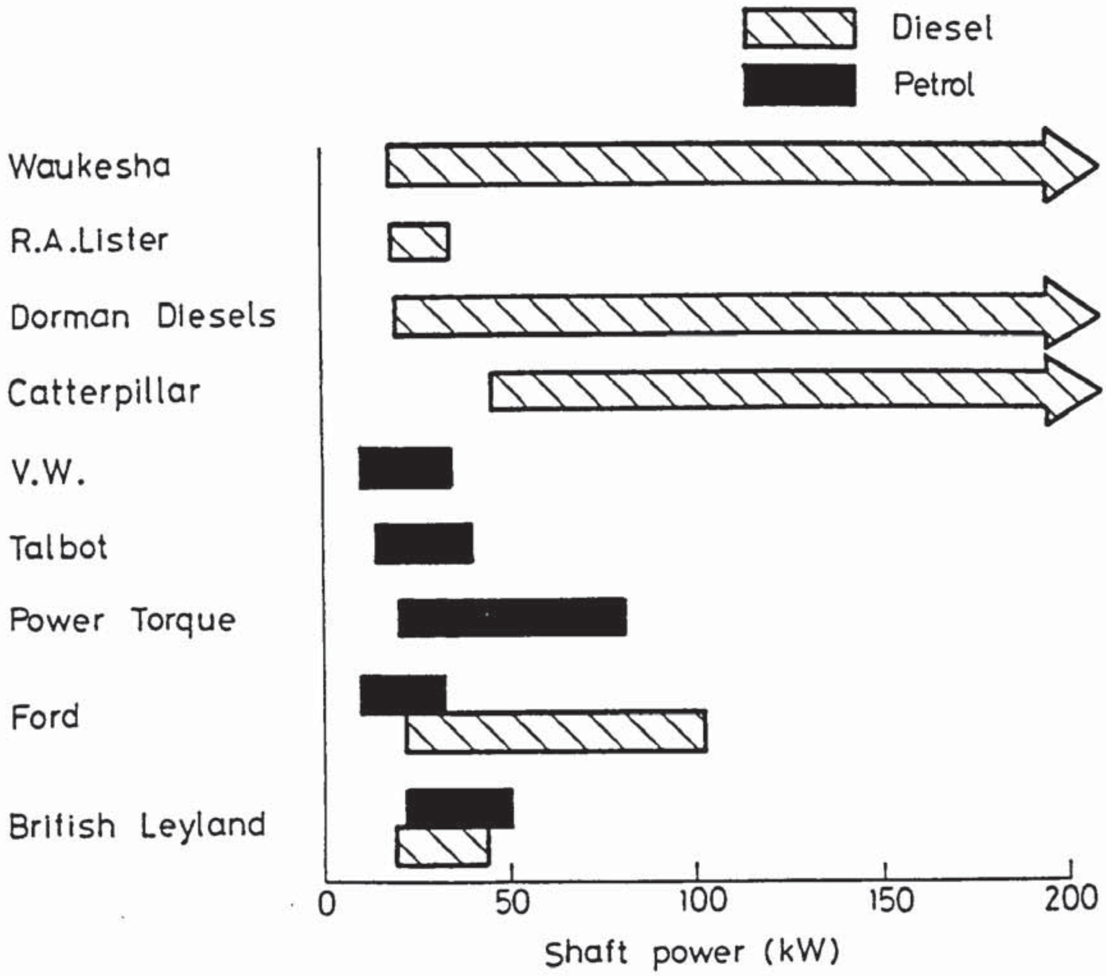


Figure 8.2 Bar Chart Of Diesel/Petrol Engines Currently Available In The U.K.

b) Only Ford/Power Torque, Dorman Diesels and Catterpillar had the ability to supply dual fuel engines.

c) The Dorman Diesel 'gas' engines are air cooled.

d) The costs of the Catterpillar engines were approximately twice the costs of the Ford/Power Torque engines.

The Ford/Power Torque engines were selected and the following engines constituting the range:

Kent 2274E	1.6 Litre	Natural Gas
S.I.4/Dover 2722	4.15 Litre	Natural Gas/Diesel
S.I.6/Dover 2725	6.22 Litre	Natural Gas/Diesel

Two major factors resulted in this range of engines being extended to include the British Leyland Land Rover power unit. The first was the necessity for dual fuel on the smallest unit. Petrol was considered unsuitable because of its storage restrictions, and the diesel based Land Rover offered the ideal solution. Secondly, at the time of formulating the range of modular units an order was received for a gas engine driven heat pump, in which the customer insisted that the power unit should be of British Leyland origin.

The gas conversion of the Land Rover engine is performed by Thornycroft Engines, and is classified as:

140G/Land Rover	2.286 Litre	Natural Gas/Diesel

8.2.2 Compressors For Modular Units

A compressor survey was made using "The Refrigerant and Air Conditioning Yearbook 1981" [99] as the source. This survey was restricted to open type screw and reciprocating compressors.

Screw compressors were considered to be preferable, and this conclusion was based upon the work of Steimle and reported by Masters [15], who suggests that the out of balance forces of a reciprocating compressor and an internal combustion engine generate excessive vibration problems. This is confirmed by Linnell [100] who suggests that a secondary flywheel, located between the engine coupling and the reciprocating compressor, is necessary to prevent premature failure of the coupling.

From this survey a range of screw compressors, produced by Bitzer Kuhlmaschinebau GmbH, was selected, which had power requirements compatible with the power developed by the engines considered in Section 8.2.1.

ENGINE	COMPRESSOR	
	MANUFACTURER	REFRIGERANT R12/R22
KENT 2274E	ROTOCOLD LTD.	AGR 450
LAND ROVER 140G	BITZER GmbH	OS 6161
DOVER S14	BITZER GmbH	OS 7051
DOVER S16	BITZER GmbH	OS 7061

Table 8.1 Engines And Compressors Used For The Range Of Modular Gas Engine Driven Heat Pumps

8.2.3 Modular Units

The next stage of the design process was to determine the ancillary equipment necessary for successful operation of the engine-compressor module. These components were similar to the equipment fitted to the prototype gas engine heat pump discussed in Chapter 3, and three main factors influenced their selection:

a) The recommendations of the engine and compressor manufacturers, based on their existing operating experience.

b) The experience gained during the development of the prototype gas engine heat pump.

c) The statutory requirements for the operation of natural gas fuelled engines [62].

Detailed performance testing was not attempted. The system performance for the full range of modular units was based on the manufacturer's published data, with modifications in accordance with the experience gained from the prototype gas engine heat pump evaluation.

8.3 PRODUCT MANUAL

As outlined in Section 8.2, the policy of Denco Air is for the engineering division to supply modular units, and for the contracting divisions to co-ordinate the overall project. Obviously technical support from the engineering division is necessary, and for the gas engine heat pumps, a product manual has been produced by the author [101]. This gives guidelines for system design around the modular unit. Besides tabulated performance data for the modules, this product manual outlines preferred installations, methods of heat exchanger and pipework sizing, expansion valve selection procedures, and building heat load calculations.

8.4 COMPUTER DESIGN AID

A computer design aid has been devised by the author to select the best module for a given installation. A flow chart and program listing for this computer design aid appears in Appendix 4. The package is designed for use with Microsoft Basic on the Mimi 803 micro computer.

The minimum requirements for operating the program are: evaporating temperature, condensing temperature, and heat load requirement. The ability to pre-set the required engine speed and to select the anticipated range of evaporating temperature exists, but if these are unknown, the program defaults to maximum engine speed and design evaporating temperature. The program operator must make the necessary allowances for supplementary heating.

The flow diagram (Figure A.4.4.1, page 278) outlines the possible program permutations. Initially there is the choice of refrigerant, which is currently limited to R12 and R22. The maximum condensing temperature for R22 is set at 55^{o}C, above which the program automatically defaults to refrigerant R12.

As the capital cost of the equipment increases through the range, the program evaluates the heat output for the smallest module first. If the output exceeds the requirement, then this is the data printed. If however the output is below the heat load requirement, then the next size module is evaluated. If none of the standard modules are capable of providing the design heat requirement, the program evaluates multiple modules to match the heat load requirement.

Programs written by others can be used to calculate the size of the requisite heat exchangers based upon the information provided by this program. However, it is suggested that in the future the fluidised bed option will exist, and at such time, design details for this should be included in the computer design aid.

8.5 HEAT PUMP TEST FACILITY

It is important to be able to commission a product prior to its despatch, and to facilitate this a closed refrigerant vapour loop test rig was designed by the author.

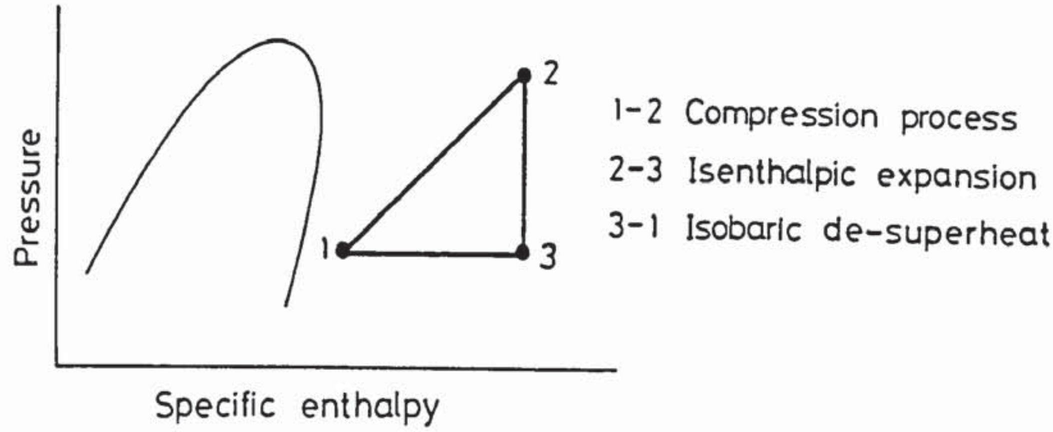


Figure 8.3 Pressure-Enthalpy Diagram For Closed Refrigerant Vapour Loop Cycle

A refrigerant vapour loop was selected because the test rig was required essentially for commissioning purposes, and not for performance testing. In a closed loop of this type the compressor discharge vapour is expanded, desuperheated, and then returned to the compressor for compression as shown in Figure 8.3.

The necessity to supply liquid refrigerant to a rotary sliding vane compressor for cooling purposes and to test the full range of gas engine heat pump modules, complicates this gas loop. However, the practical solution is to use two desuperheaters. These are linked in series for the largest module and in parallel for the module incorporating the AGR 450 rotary compressor.

A schematic diagram of this system is shown in Figure 8.4, and the principle of the operation is given in Appendix 5.

8.6 MAINTENANCE

Internal combustion engines are notorious for their need for frequent servicing. However, by following the recommendations of Pegley et. al. [14], outlined in

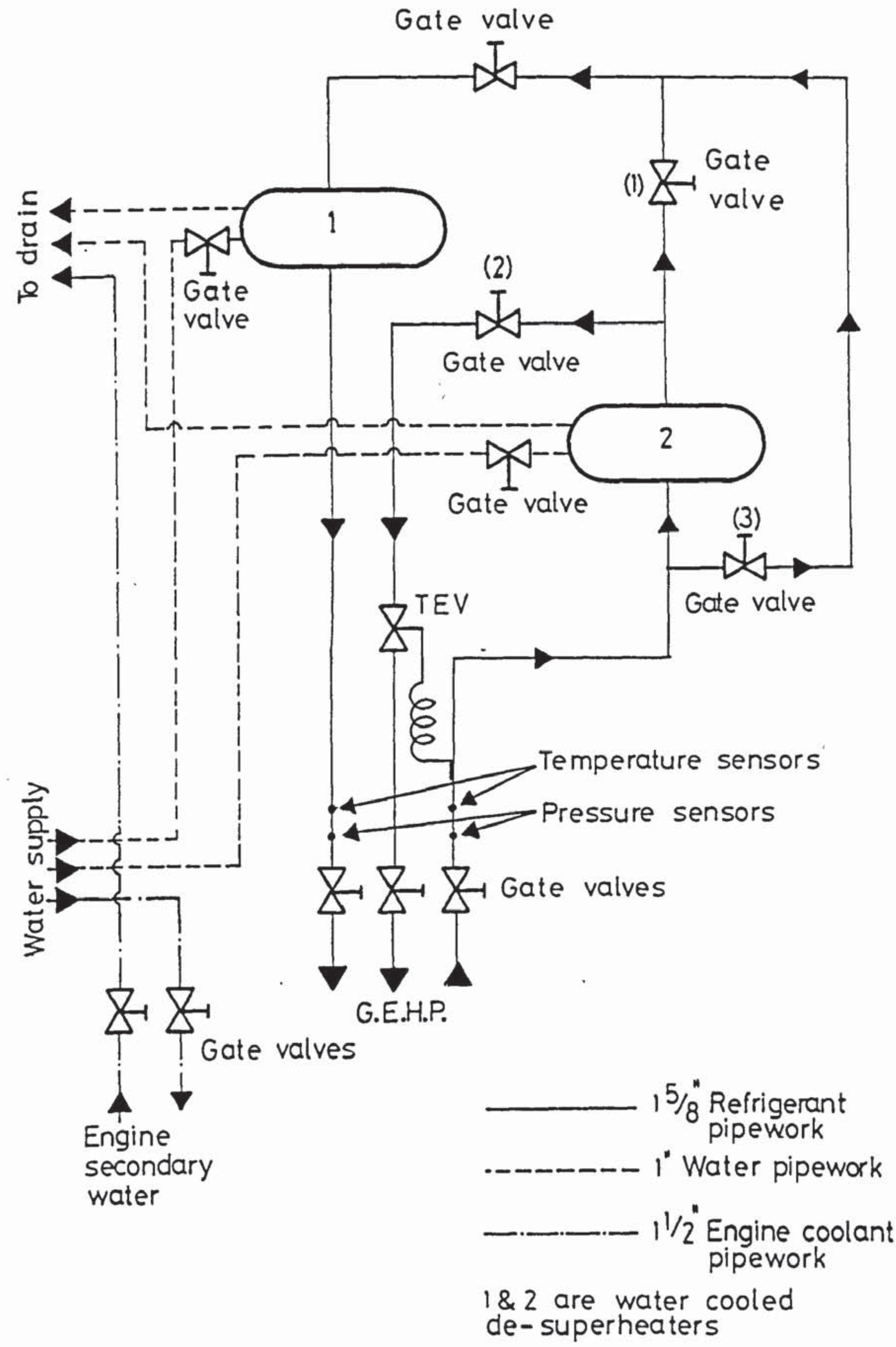


Figure 8.4 Schematic Arrangement G.E.H.P. Test Facility

Chapter 2, the service interval for the gas engine heat pump system was extended to 1000 hours.
The condition of the oil is the limiting factor of this service interval, and so the use of an oil analysis service, such as offered by Shell Oil Care is recommended. This service requires that a monthly oil sample from the power unit is dispatched to Shell Research Laboratories, and a diagnostic report for the sample is made by return of post. If a condition requiring attention is observed, this report is preceded by telex or telephone notification. The experience gained from an analysis such as this may enable the service interval to be further extended.

Complex plant, such as gas engine driven heat pumps sometimes develop minor faults which can result in excessive non operating time. It is therefore necessary for service personnel to be readily available. Denco Air has an excellent service division for refrigeration and air conditioning, but the personnel were unfamiliar with internal combustion engine fault diagnosis. Although the engine suppliers were prepared to offer service contracts for the engines, they were not conversant with refrigeration techniques. This would have resulted in two service technicians attending a site for each breakdown and regular service. Thus it was decided to train the Denco service personnel in internal combustion engine maintenance. An internal publication entitled "Installation and Maintenance Manual For Gas Engine Driven

Heat Pumps", was prepared by Hickman and Watkins [63], based upon workshop manuals for the range of engines and compressors, and on experience gained by the authors during the course of product development. A small team from the service division was selected to maintain the gas engine driven heat pumps, and members have attended a number of short courses designed in conjunction with the engine manufacturers to give a broad understanding of internal combustion engine servicing and troubleshooting. This approach lead to additional problems initially. However, with the assistance of the Denco Air development engineers, and the engine manufacturers the early inhibitions have been surmounted, and very good service back-up now exists.

8.7 RELIABILITY

Reliability is a major factor influencing the sale of gas engine heat pumps, and so this Section attempts to quantify system reliability. Breakdowns which have occurred, particularly with the early units, can be regarded as operating experience. Some component items have been found to be unsuitable for applications with long annual running hours and have been replaced, whilst other components have been incorrectly applied. A complete breakdown of the faults experienced to date and the action taken to prevent their repetition, is contained in Appendix 6. It can be seen that the majority of faults have occurred on a one-off basis and, although breakdowns will never be completely eliminated, appropriate remedial action has been taken to prevent the repetition of nuisance failures.

8.8 INDUSTRIAL AND COMMERCIAL APPLICATIONS FOR HEAT PUMPS

Numerous potential applications exist for heat pumps, and these are outlined below.

8.8.1 Space Heating

Heat pumps for space heating are mainly air source, and have been developed as packaged units capable of producing warm air or hot water, and it is suggested by Masters et. al. [46] that engine driven heat pumps are capable of displacing conventional heat generators once they have gained market acceptance. Kalisher [102] suggests that the current lack of market penetration is due to:

a) Energy savings are not highly valued by the individual.

b) There is a general lack of understanding of the technology.

c) Payback periods are often excessive due to the low annual operating hours.

8.8.2 Swimming Pool Applications

Heat pumps are ideally suited to swimming pool applications for two main reasons:

a) Pool water and air temperatures of 27°C and 28°C respectively are normally required which means that year round heating is necessary in the British Isles, and these high operating hours considerably reduce heat pump payback periods.

b) Because of water evaporation from the pool, high humidity levels are present in pool halls. To prevent structural damage, humidity must be controlled, and it is suggested by Masters et. al. [46] that air changes in excess of 10 per hour are necessary.

According to Reay and MacMicheal [11] almost 60% of the heat loss from swimming pools is due to ventilation losses, and although conventional heat recovery equipment can recover the sensible heat, the latent heat of the water vapour is lost. A heat pump will recover this latent heat, and if ozone is used to disinfect the pool, the dried ventilated air can also be recirculated.

8.8.3 Drying

Hodgett [103] suggests that heat pumps could well supplement conventional gas-gas waste heat recovery in drying applications, because of the high energy savings possible. Reay [104] claims that a better quality product is obtained using a heat pump, since the atmosphere is more closely controlled. However, the efficiency of many existing systems is questioned by Hodgett [103] since it is not universally appreciated that the system must be thermally well insulated and vapour tight. Hodgett [103] reports that a 5% air leak results in up to 25% loss of thermal efficiency.

8.8.4 Evaporation And Distillation

There are three applications for this type of process:

i) Produce concentration.

ii) Volume reduction of liquid effluents.

iii) Recovery of water (generally the condensate) for re-use.

Besides the conventional apparatus Reay [105] outlines two types of heat pump system that can be utilised.

a) Indirect Mechanical Vapour Recompression

This incorporates a conventional heat pump circuit. The product of distillation is passed over the evaporator where it condenses giving up its latent heat to the refrigerant. The refrigerant is then compressed, and passed through the condenser, which is located in the liquid being distilled.

b) Direct Mechanical Vapour Recompression

The vapour from the distillation column is used as the working fluid. It is compressed, and then passed through the condenser located in the liquid, and has the added advantage of a lower compression ratio than the indirect system. For distillation the vapour is rarely water, and in order to utilise the direct method the compressor must be capable of handling the product of distillation.

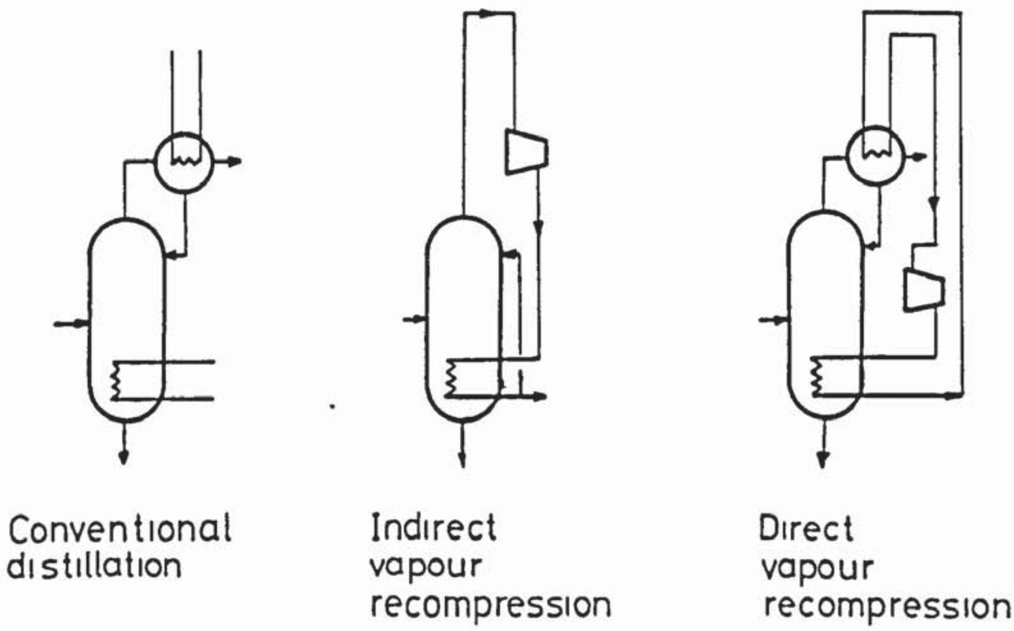


Figure 8.5 Distillation Techniques

8.8.5 Heat Recovery From Liquid Effluents

McMullen et. al. [4] suggest that heat pumps are only practical for sensible heat recovery when the temperature of the effluent is below a usable level, since conventional heat recovery equipment will recover high temperature sensible heat more economically.

8.8.6 Heat Recovery From Refrigeration Plant

Traditionally the heat rejected from refrigeration plant is degraded to ambient temperatures either directly, or by cooling towers, but there are many applications where Reay et. al. [11] suggest that it would be worthwhile utilising this rejected heat if only to preheat a process fluid.

A comprehensive matrix of possible industrial and commercial heat pump applications is shown in Figure 8.6. However, heat pumps must compete commercially with conventional heating systems which generally have lower capital costs, so it must be stressed that system economics are critical.

PROCESS \ INDUSTRY	Dairy	Grain Mills	Lumber & Wood	Leisure Centres	Distillery	Chemical Plants	Food Processing	Confectionary	Textiles	Petrol Refining	Mechanical Engineering	Electrical Engineering	Hotels	Shops & Stores	Greenhouses	Farming
Washing	X	X					X	X					X			X
Cooking							X	X					X			
Pasteurising	X															
Pool Heating				X												
Evaporation						X	X									
Propagation															X	X
Drying		X	X						X							
Distillation					X	X										
Sterilisation	X						X									X
Space Heating	X			X		X	X				X	X	X	X	X	X
Heat Recovery	X										X					
Pressing									X							
Vessel Pre-Heat					X	X				X						
Process Heat					X	X	X	X	X	X	X	X				
Refrigeration	X			X			X					X	X	X		X
Domestic Hot Water	X			X			X	X			X	X	X	X	X	
Dehumidification		X	X	X							X	X	X			

Figure 8.6 Matrix Of Possible Heat Pump Applications

8.9 ECONOMICS

The capital cost of an electrically driven heat pump is significantly higher than that of a conventional boiler system with a similar heat output. An engine drive for a similar heat pump increases the cost still further. This is due to the inherently higher cost of internal combustion engines compared with electric motors, together with the cost of ancillary equipment essential to control the engine, and to recover its waste heat. Masters et. al. [46] consider the running costs per kWh for an electric heat pump, and a gas engine driven heat pump. These are compared with the running costs of a gas boiler system having a gross efficiency of 75%. If the fuel costs used in this analysis are updated in line with inflation (see Table 8.2), the following conclusions may be drawn:

a) For a temperature difference between the heat source and the heat load greater than 40K, the electric heat pump will never have a lower operating cost than the conventional gas boiler system.

b) The gas engine driven heat pump has a lower operating cost than the electric heat pump and the conventional boiler system, but as the temperature differential between the heat source and the heat load increases, the cost savings are reduced.

c) The economics of each individual installation must be considered in depth before the decision to install a heat pump is taken.

The economic analysis most frequently adopted is the payback period, which is defined as:

$$\text{SIMPLE PAYBACK} = \frac{\text{Increase in Capital Cost for Heat Pump Installation}}{\text{Annual Fuel Cost Savings}} \qquad (8.1)$$

Manion [107] suggests that the results of this simple payback calculation are readily accepted, because they are easily understood. The cost savings are however generally based on steady state performance characteristics which neglect the effect of frost formation. Additionally the electricity consumed by ancillary equipment such as water pumps, fans and control systems are often omitted from such calculations. If interest charges and inflation are also omitted then payback can only be used as a guide.

Energy Form	Unit of Supply	Average Price per Unit of Supply (1983)	Average Price Pence per gross kWh (1983)	Increase in cost 1982-83
ELECTRICITY				
Direct	kWh	3.74p	3.74	10.3%
Off Peak	kWh	1.67p	1.67	9.1%
NATURAL GAS	ft^3	0.314p	1.04	NIL
FUEL OIL				
General Zone 35 secs	Litre	23.90p	2.26	22.6%
General Zone 200 secs	Litre	20.47p	1.82	15.6%
General Zone 950 secs	Litre	18.05p	1.60	15.8%
General Zone 3500 secs	Litre	17.14p	1.50	16.6%
PROPANE	Tonne	£238.10	1.70	11.8%
BUTANE	Tonne	£214.00	1.55	22.3%
COAL	Tonne	£57.30	0.73	7.7%
INDUSTRIAL COKE	Tonne	£84.30	1.09	NIL

Table 8.2 Fuel Cost Comparison [106]

Numerous proposals for a more acceptable analysis have been reported in the literature, and a payback period evaluation suitable for gas engine heat pump installations which has been based on the suggestions of authors in references [38,107,108,109,110,111,112] is given in Appendix 2.

Other factors may also affect the economic viability of heat pumps. In the early 1950s John Laing Ltd. developed a heat pump for domestic heating. However, the government of the day decided that a heat pump was a luxury and applied a 100% purchase tax effectively destroying the economics. A few years later Lucas developed a similar domestic heat pump. Unfortunately at about this time the Electricity Council introduced low off-peak tariffs to encourage the use of night storage heaters, which considerably affected the Lucas market, resulting in the product being dropped [113].

The use of off-peak elecrtricity to drive an electric heat pump with a heat storage facility appears attractive. However, work at Cranfield Institute of Technology by Buick [114] indicates that an economic storage facility would be unable to store sufficient heat to last throughout the day, and the need to top up using peak electricity destroys the otherwise economic advantage. The advent of latent heat stores may resolve this problem.

8.9.1 Economics Of Exhaust Gas Latent Heat Recovery

For the prototype gas engine heat pump described in this work, the latent heat content of the exhaust gas cooled to 15^oC is 9.6% of the heat content of the fuel supplied to the engine. This is equivalent to approximately 10kW at the full load operating condition. As discussed in Chapter 6, to recover this latent heat a run-around coil system would be necessary. The cost for such a system would be approximately £1000-£1200, which is equivalent to £100-£120/kW output. This is excessive when compared with the figure of £80/kW suggested by Masters et. al. [46] for heat pump systems to give a payback of less than 4 years. Since heat at 15^oC is too low even for space heating applications the actual heat recovered would be less than 10kW. In addition because of the possibility of corrosion by the products of combustion, one heat exchanger in the run-around coil system would need to be constructed of a high quality stainless steel. The overall effect would be an increase in the cost per kW. Thus it is concluded that the exhaust gas latent heat could not be recovered economically.

8.9.2 Economic Performance Of Gas Engine Driven Heat Pumps

Currently 10 installations using the Denco range of gas engine driven heat pumps exist, and these are installed for either space heating or swimming pool applications. Annual fuel cost savings of more than £17,500 has been quoted in "Energy Manager" [115] for a swimming pool application utilising two Denco gas engine driven heat pumps, and savings in excess of £30,000 have been achieved for a space heating application in less than two years. Both of these installations have won gas energy management awards (GEM).

8.10 SUMMARY

An increasing market has been identified by Denco Air Limited for gas engine driven heat pumps.

The following technical support has been provided in an effort to penetrate this market.

a) A range of modular gas engine driven heat pumps have been devised.

b) A product manual and computerised unit selection aid have been prepared.

c) Development of a prototype unit has resulted in an extended service interval of 1000 hours, and it is suggested that by utilising an oil analysis service, the service interval may be further extended.

d) Training has been provided to ensure that efficient maintenance support exists.

e) A heat pump test facility allows for full commissioning prior to despatch.

Plant reliability has been greatly improved enabling the Denco sales division to penetrate the market, particularly for space heating and swimming pool applications. A concerted effort is now required by the sales division to consolidate the Denco position as market leader, and a breakthrough in some of the application areas outlined in Figure 8.8 is required.

Applications in dairies, distillation and drying plants appear to be the most suitable economically. However, the economics of each proposed installation must be considered to ensure viability of the project. Although simple payback calculations give a good indication, an indepth analysis based upon seasonal performance, life expectancy, interest charges and inflation is necessary.

CHAPTER NINE

CASE STUDY : GAS ENGINE DRIVEN HEAT PUMP FOR A SWIMMING POOL APPLICATION

9.1 INTRODUCTION

The following is a case study of the first commercial gas engine driven heat pump installed by Denco Air Limited. It is by no means a typical installation, but it is important because of the experience it provided.

The study is divided into two parts:

i) application experience;

ii) system performance.

9.2 APPLICATION EXPERIENCE

During this first year of operation numerous problems developed that were not envisaged by the contracting division at the projecting stage.

This Section highlights those problems, and the remedial action taken, largely by the author.

a) Commissioning:

The installation was commissioned in January 1982, and a schematic diagram of the plant is shown in Figure 9.1.

During commissioning, it was found that if all the pool water flowed through the calorifier (i.e. valve 'A' open 1-3), the pressure drop through the calorifier resulted in a reduction of pool water flow rate below that necessary to maintain adequate chlorination. Hence valve 'A' had to be set manually to give partial flow in direction 2-3. Consequently air heating alone could not be achieved.

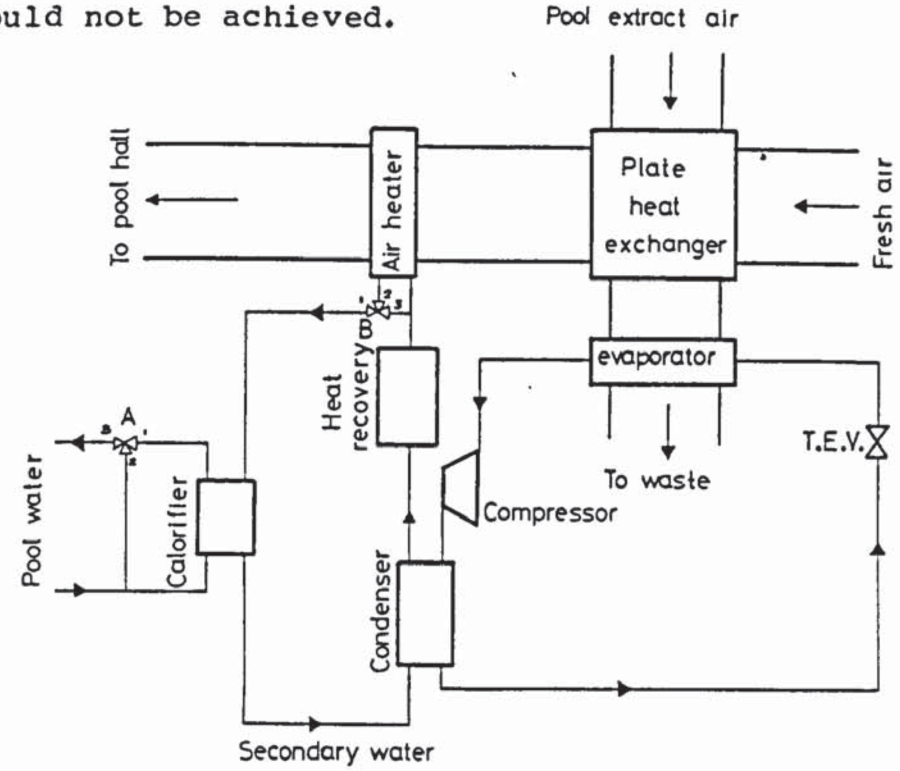


Figure 9.1 **Schematic G.E.H.P. Installed For A Swimming Pool Application. A And B Are Electrically Controlled Modulating Three Way Valves.**

b) Cut Out Due To High Compressor Discharge Pressure:

During the spring of 1982 the heat pump suffered numerous high pressure cut outs during night time operation. The high pressure switch was set at 20.5 bar, and during the daytime operation the compressor discharge pressure never exceeded 18.5 bar. These cut outs were curious, in that the ambient temperature fell during the night. The obvious result of a lower ambient temperature would be a reduction in the evaporating pressure, and hence the condensing pressure. However, as the ambient temperature fell, the demand for pool water heating increased. The controller was programmed to give pool water heating priority, hence valve 'B' (Figure 9.1) modulated to give flow in direction 3-1 bypassing the air heater. Because valve 'A' was partially open 2-3, insufficient cooling water was supplied to the calorifier. The temperature of the secondary water returning to the condenser increased, with a subsequent increase in condensing pressure, until cut out occurred.

The short term solution to this problem was to reduce the engine operating speed during the night. In the long term a booster pool water pump was necessary.

c) Excessive Pool Water Temperatures:

In the summer of 1982, excessive pool water temperatures were experienced, whilst the air temperature never reached the set point. Observation of the control system indicated that as the air temperature approached the set point, valve 'B' (Figure 9.1) slowly modulated to give increased flow in direction 3-1. The majority of heat was therefore passed to the calorifier, and because of the manual override on valve 'A', heat was dumped into the pool. It should be noted that if valve 'A' had not been manually set, the design control sequence would have resulted in system lockout.

In an effort to overcome the problem the following simulation was attempted:

i) Valve 'A' was regulated to give no pool water flow through the calorifier.

ii) Valve 'B' was regulated to give full secondary water flow through the air heater.

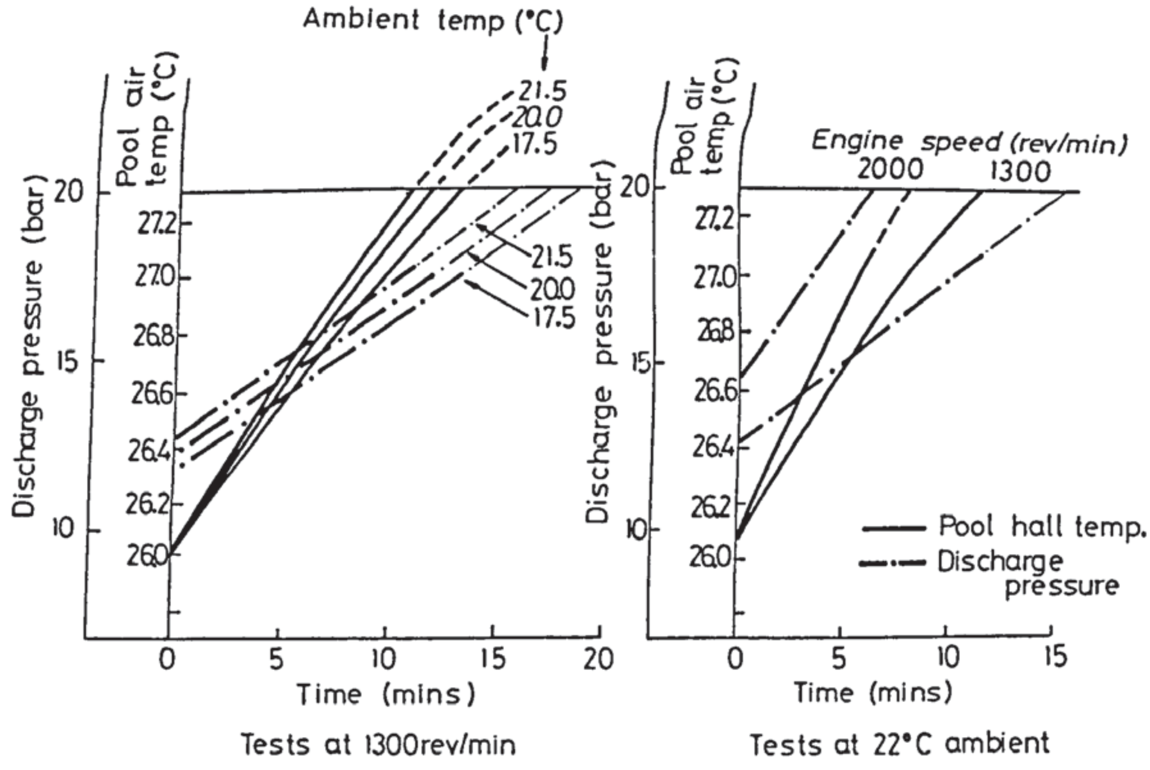


Figure 9.2 High Discharge Pressure Lockout Simulations

At all engine speeds above 1500 rev/min high discharge pressure lockout occurred before the pool air set point temperature was reached.

Below 1500 rev/min the pool air reached set point temperature before high pressure lockout occurred. Results of this simulation are shown in Figure 9.2.

The plant was allowed to operate under these conditions for an eight hour test period, during which time the ambient temperature increased from 17.5°C to 21.5°C. During this test period the total engine operating hours were 0.9h, compared with continuous operation during the previous 24 hour period. A gradual decline in the pool water temperature was also observed.

To overcome the problem of excessive pool water temperature the following recommendations were made:

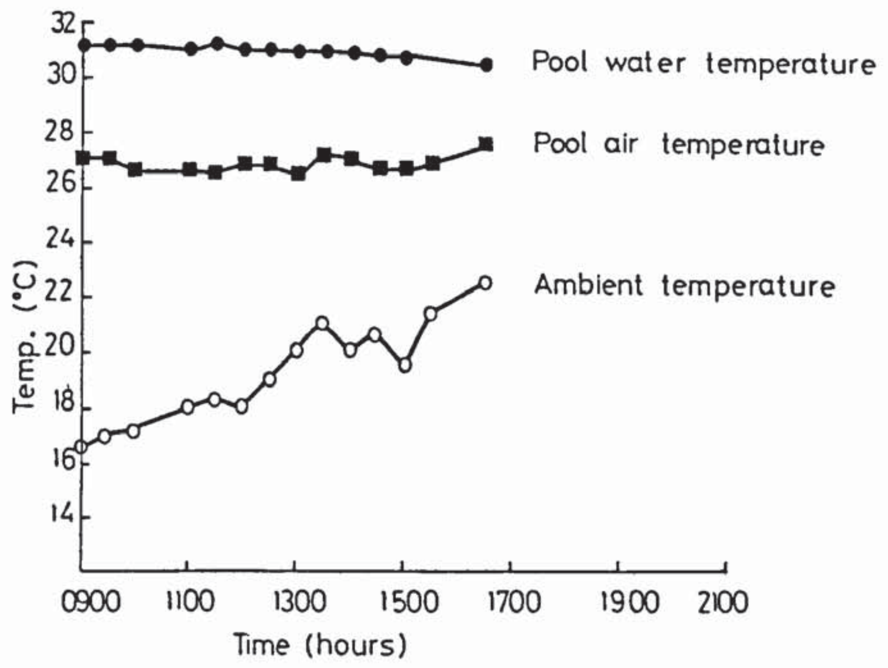


Figure 9.3 G.E.H.P. Plant Characteristics During Simulations Tests

i) Increase the size of the water to air heat exchanger.

ii) Modify the control system.

The recommendation to increase the size of the water to air heat exchanger was rejected by the Denco management, thus the control system shown in Figure 9.4 was proposed. The control sequence is tabulated in Appendix 5.4.

Because the size of the air heater was not increased, high pressure cut out was still possible in the air heating mode. To avoid this a compressor discharge pressure sensor was proposed. This operated in two stages:

i) The engine speed was reduced if the compressor discharge pressure rose above 18.5 bar.

ii) The three way valve B opened to give flow in direction 1-2 on a rise in compressor discharge pressure above 19.5 bar.

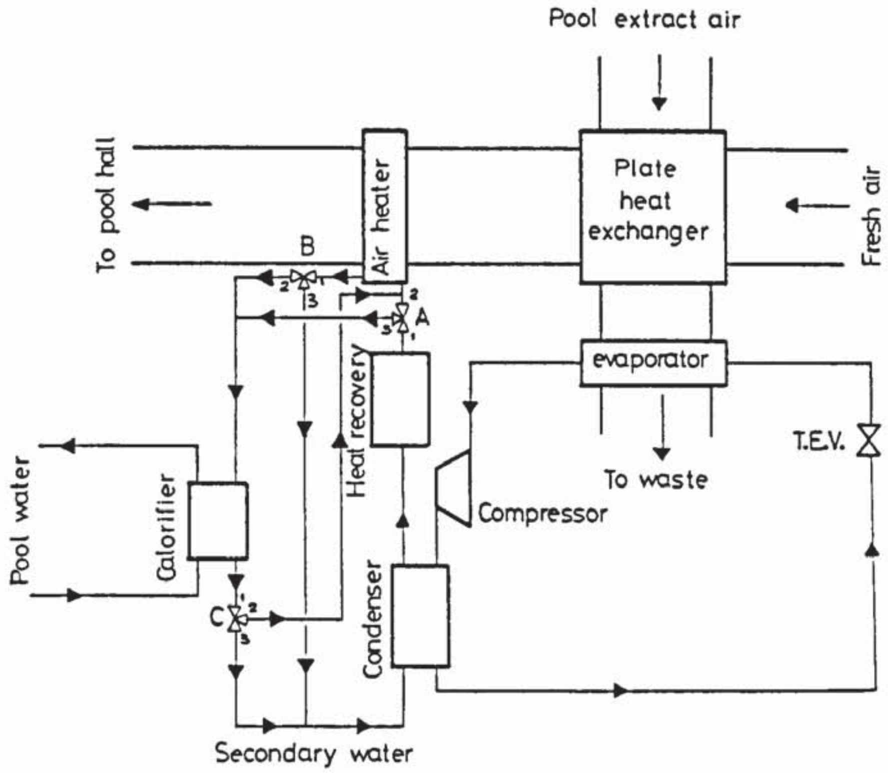


Figure 9.4 Proposed Modifications To G.E.H.P. Control System. The Three Way Valves Are Either/Or, Not Modulating.

This system eliminated the need for a modulating valve in the pool water circuit and was implemented in the autumn of 1982.

d) Insufficient Plant Capacity:

During the winter of 1982/3, the heat pump was unable to maintain the design conditions. The building heat load exceeded the heat output of the plant when running at maximum speed. An error was detected in the building heat load calcuations, indicating the need for supplementary heating. A boiler backup was then provided. Two boilers were installed in series with the condenser (see Figure 9.5).

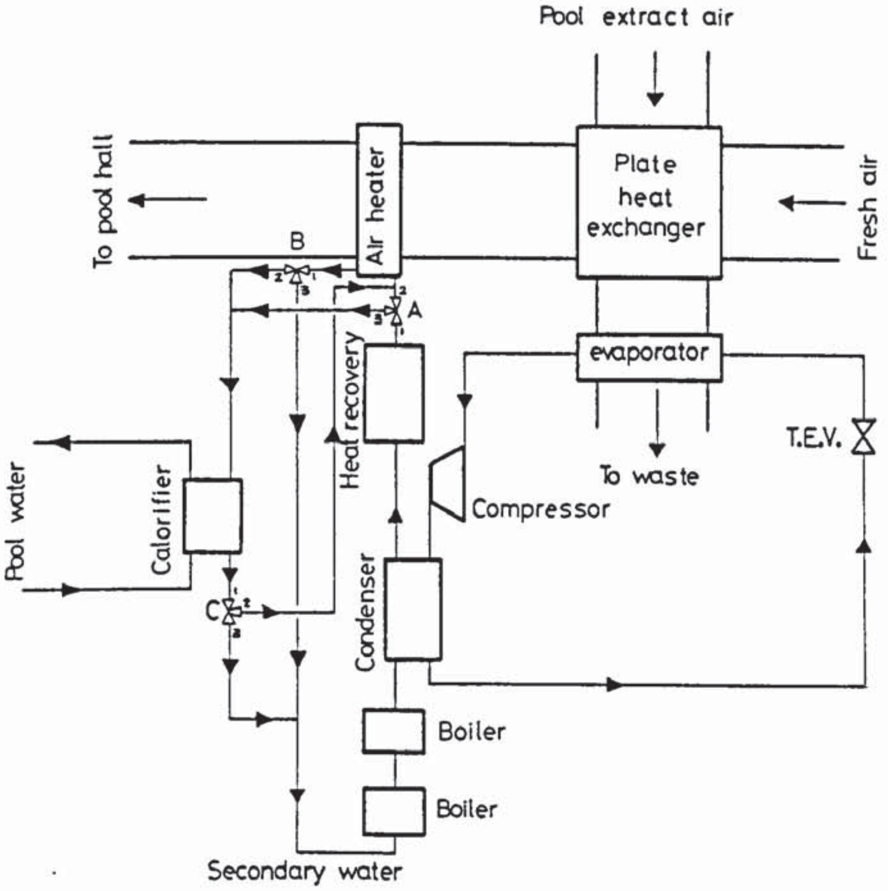


Figure 9.5 G.E.H.P. Installation With Boiler Backup

One boiler rated at 50 kW was used to boost the gas engine driven heat pump during cold weather, and was controlled by a "frostat" set at 2°C ambient temperature. The boiler thermostat was set at 35°C ± 1K to ensure that the return water temperature to the condenser did not cause a high discharge pressure. The second boiler rated at 100 kW was for standby purposes in the event of a heat pump malfunction.

e) Noise Considerations:

Complaints were received by the local Environmental Health Department concerning noise from the plant. These complaints were from people living over 100 yards from the plant room, beyond a busy railway line.

Discussions with the Environmental Health Department indicated that in the complainants' gardens the noise level was approximately 32dBA compared with a background noise level of 25-28dBA. (Readings taken at 9 p.m.).

The Environmental Health Department's requirements were met by installing noise attenuation equipment which reduced the noise level by 4dBA at 3m from the plant room. A full spectrum analysis is contained in Appendix 3.

9.3 SYSTEM PERFORMANCE

Performance characteristics during the first year have been ignored, due to the operational difficulties reported in Section 9.2. However for the second year of operation, system performance has been measured by a heat meter, and a fuel consumption monitor.

Readings from both the gas meter, and the heat meter were made on a monthly basis, and used to calculate the average heat output, the fuel consumption, and the primary energy ratio.

Details for the twelve month period April 1983 to April 1984 are shown in Figure 9.6.

The maximum primary energy ratio obtained during this period was 1.63 the minimum being 1.02. A seasonal primary energy ratio has been calculated as 1.13, and the annual fuel savings based upon a fuel cost of 1.04p/kWh compared with a gas boiler of 70% thermal efficiency have been found to be over £3500 (see Appendix 2).

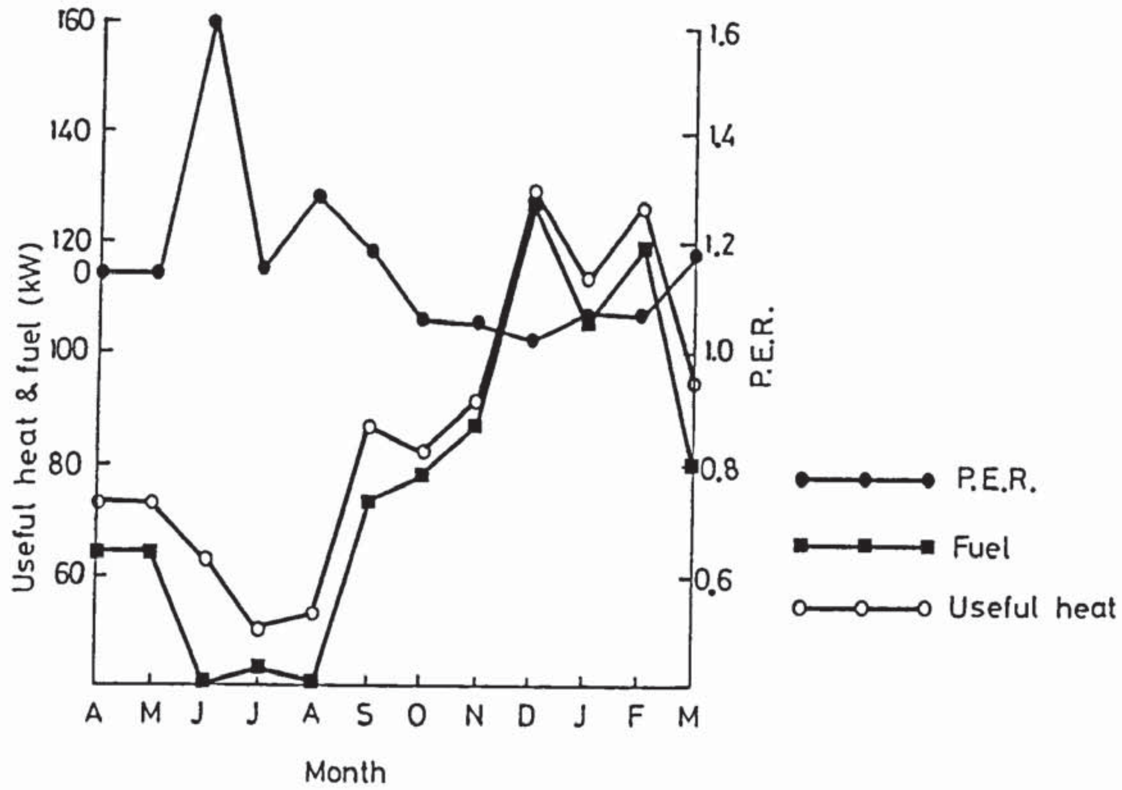


Figure 9.6 **Performance Characteristics: G.E.H.P. Installation With Supplementary Boiler Heating (April 1983-April 1984)**

It should be noted that these figures include the boiler contribution to the plant, and the low monthly values of P.E.R. are due to the necessity to run the boilers for either boost heat, or because the heat pump was not operational. The supplementary heating provided by the boilers was controlled by an outdoor thermostat, as recommended by Goodall [47,116] to optimise overall efficiency, which was arranged such that the boiler would not operate unless the ambient temperature was below 2°C, or unless there was a heat pump malfunction.

In addition to the fuel cost savings the gas engine driven heat pump provides useful dehumidification of the pool hall, the performance of which is difficult to quantify. However, before the heat pump was installed, dehumidification was achieved by means of roof mounted extraction fans which were very inefficient. The improvement in humidity control can be seen by comparing Figures 9.7 and 9.8.

The humidity level is maintained at 30-40% R.H. which is low. Milbank [117] summarises the relative humidity for seven local authority pools, which have an average of 57% R.H. and suggests 60-70% R.H. is ideal for comfort conditions. Reducing the air changes for the installation discussed in this case study would not be detrimental to the building structure, and would improve the seasonal performance. In addition, it would be more comfortable for the bathers since evaporation from the body would be reduced.

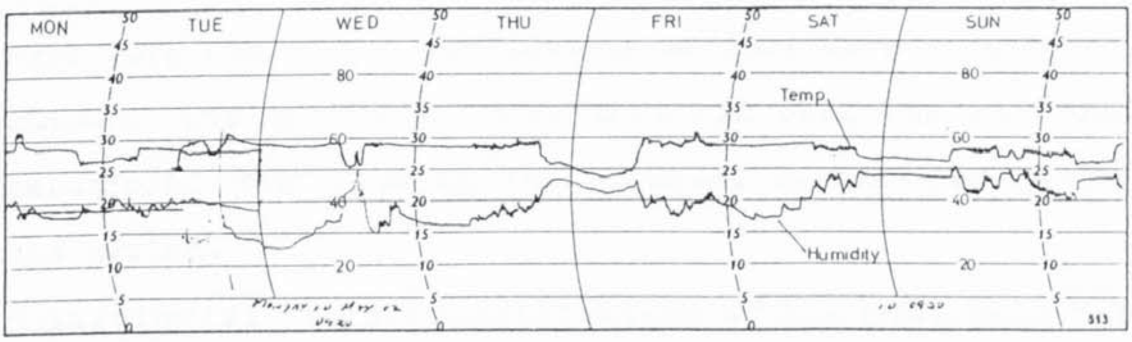


Figure 9.7 Temperature And Humidity Levels In A Swimming Pool Hall Using A Gas Engine Driven Heat Pump

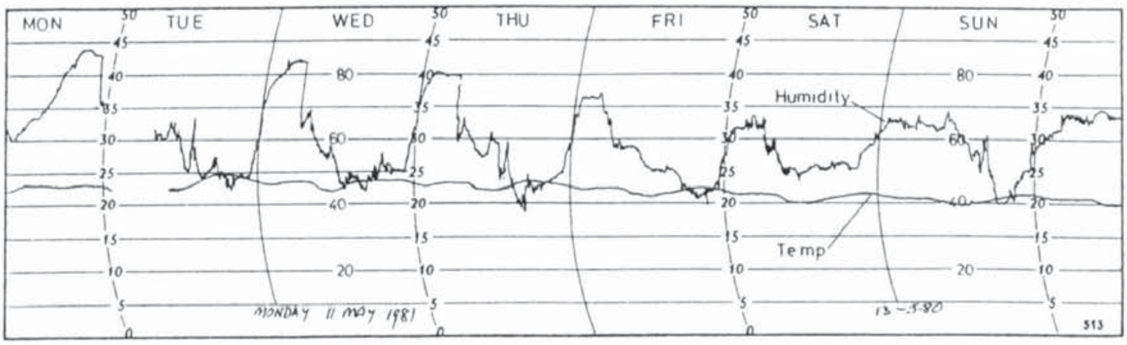


Figure 9.8 Temperature And Humidity Levels In A Swimming Pool Hall Using Gas Boilers And Roof Mounted Extraction Fans

9.4 SUMMARY

Although there have been numerous problems with the system, it was purchased knowing that this was the first swimming pool application installed by Denco Air Limited. During the first twelve months that the unit was installed there were times when replacement boilers were considered. However, the energy savings obtained when the unit was operational far outweigh the problems encountered during this period.

In particular the dehumidification of the pool hall has ensured the future of the building which prior to the installation of the gas engine driven heat pump was rapidly deteriorating due to the effects of condensation. The early problems associated with the unit now appear to be surmounted and a successful future is now anticipated. These comments echo the views of the Clerk of the Works, controlling this installation.

CHAPTER TEN

CONCLUSIONS

10.1 INTRODUCTION

This thesis has described the development of a range of gas engine driven heat pumps from their conception to commercialisation. In addition, a device to prevent the build up of frost on the surface of air heated evaporators has been proposed, and successfully tested.

The performance of rotary sliding vane compressors, as used throughout the project, has been criticised, particularly by Marquand [77] on the basis of their coefficient of performance, and by Hughes et. al. [54,58] because of the suggested effects of lubrication on performance. However, it is shown in this work that these criticisms are unfounded.

The achievements of the project are reviewed in this chapter, which is divided into four parts:

a) Development of the Gas Engine Driven Heat Pump

b) Rotary Sliding Vane Compressor Performance

c) The Fluidised Bed Defrost Mechanism

d) Commercial Considerations

10.2 DEVELOPMENT OF THE GAS ENGINE HEAT PUMP

The steady state performance of a prototype gas driven heat pump was found to be comparable with the published data for the engine and the compressor. However, because of inefficiencies associated with the installation, in particular the engine waste heat recovery equipment, and the refrigerant evaporator, the total system output, and hence primary energy ratio were not optimised.

There is a marked reduction in condenser heat rejected with time, on the prototype gas engine heat pump. This is due to the combined effects of a high ambient air temperature approach to the evaporator and a reduction in volumetric air flow rate, with increased air side resistance due to frost formation. Increasing the volumetric air flow, by the use of centrifugal fans, and the size of the evaporator would reduce this temperature approach. Centrifugal fans would also be necessary to overcome the increased resistance due to frost formation.

Engine waste heat recovery can be improved using additional heat exchangers, however the latent heat content of the exhaust gases cannot be recovered economically. This is because complete combustion of the fuel cannot be guaranteed, and a run-around coil system is necessary to recover this latent heat.

Other system inefficiencies are due to convection and radiation losses from the engine-compressor module. Positioning the evaporator remote from the module would eliminate forced convection losses, and natural convection could be reduced by sealing the engine enclosure.

10.3 COMPRESSOR PERFORMANCE

A discrepancy between the measured power absorbed by the compressor, and the calculated work of compression based upon the refrigerant properties, and thermodynamic considerations was observed. It is concluded that this was due to effects of liquid refrigerant injection into the compressor for cooling purposes and not due to the effects of lubrication as suggested by Hughes et. al. [54,58]. This liquid refrigerant injection results in an increased mass flow rate at compressor discharge compared with compressor suction. Little immediate effect on compressor performance was observed when this liquid refrigerant injection was suppressed for a short period. However, long term degradation of the compressor vanes would occur if liquid refrigerant were removed for prolonged periods, and would ultimately result in a loss of compression.

The full load Coefficient of Performance of the compressor is less than that of an equivalent reciprocating machine. However, when used for heat pump applications the seasonal C.O.P for both machines would be expected to be comparable. This is because of the necessity in practice to off-load cylinders on the reciprocating compressor, as the heat demand falls with rising ambient temperatures.

10.4 FLUIDISED BED DEFROSTING MECHANISM

A fluidised bed defrosting mechanism has been successfully tested. This approach to the defrosting problem is superior to conventional defrosting techniques since energy is not wasted during the defrosting process. Some additional energy is necessary due to the increased air pressure drop associated with this type of reactor, but the performance of the evaporator is improved using a fluidised bed because of an increase in the overall heat transfer coefficient.

The net result is an improvement in the primary energy ratio, and the elimination of traditional defrost cycling. Further development work is now necessary to perfect the technique of fluidisation.

10.5 COMMERCIAL CONSIDERATIONS

Extensive technical support has been provided in an effort to attain the gas engine heat pump market share anticipated by Denco Air Limited. However, a concerted sales effort is now required to consolidate the position as a market leader.

A unit capable of providing heat at temperatures in excess of 100°C would aid this market expansion. However, consideration of the economics of each installation must be made to ensure project viability.

The first commercial application of the gas engine heat pump provided considerable operating experience, but it was apparent that:

a) The contracting divisions at Denco Air had insufficient expertise in control system design. As a result Denco Magna, a subsidiary within the Denco Group were commissioned to provide the specialist expertise on all future installations.

b) Insufficient data was available to the contracting division, so a manual was prepared by the author.

c) All installations required some form of supplementary heating, and so gas boiler systems were adopted.

Considerable energy savings are obtained when using gas engine heat pump systems in certain applications. However, the necessity to provide supplementary heating erodes some of these savings, and together with increased capital costs effectively increases the payback period. The use of dual gas engine heat pumps may further increase capital costs, but, additional energy savings may result in a lower payback period. It is concluded that this option should be considered for each future project.

CHAPTER ELEVEN

RECOMMENDATIONS

The following recommendations are made, based upon the work reported in this thesis:

a) Initially, the size of the refrigerant evaporators should be increased and the evaporators repositioned remote from the engine compressor module. Air flow rates should also be increased, and centrifugal fans adopted. Further development work on the fluidised bed defrosting mechanism is necessary, and when complete, fluidised bed evaporators should replace the conventional evaporators.

b) Engine heat recovery using Bowman heat recovery equipment is recommended but a secondary exhaust gas heat exchanger should be fitted to all future installations. Recovering the exhaust gas latent heat is not recommended.

c) Removing the electronic equipment and the gas train from within the engine-compressor module should be considered. This will allow the enclosure to be sealed, and hence reduce natural convection heat losses.

d) A Shell oil care analysis for the engine should be initiated to determine if a further increase in the service interval is possible.

e) To consolidate the position as a market leader, sales effort in the following areas is necessary:

   i) Space Heating

   ii) Leisure Industries

   iii) Dairies

   iv) Distillation

   v) Drying

f) Due to the high capital cost of gas engine heat pump systems, it may be pertinent to lease units in the present economic climate.

g) The development of a unit capable of supplying heat in excess of 100°C is essential. The work by I.R.D. [29,33] would be a good basis for this development.

h) Dual engine driven Units, giving additional capacity control should be considered at the estimating stage.

i) The Rotary Sliding Vane Compressors should not be operated for extended periods without liquid injection.

j) The experience obtained from gas engine driven heat pump systems should now be used to develop a range of Combined Heat and Power Units.

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APPENDIX 1

PHOTOGRAPHIC PLATES

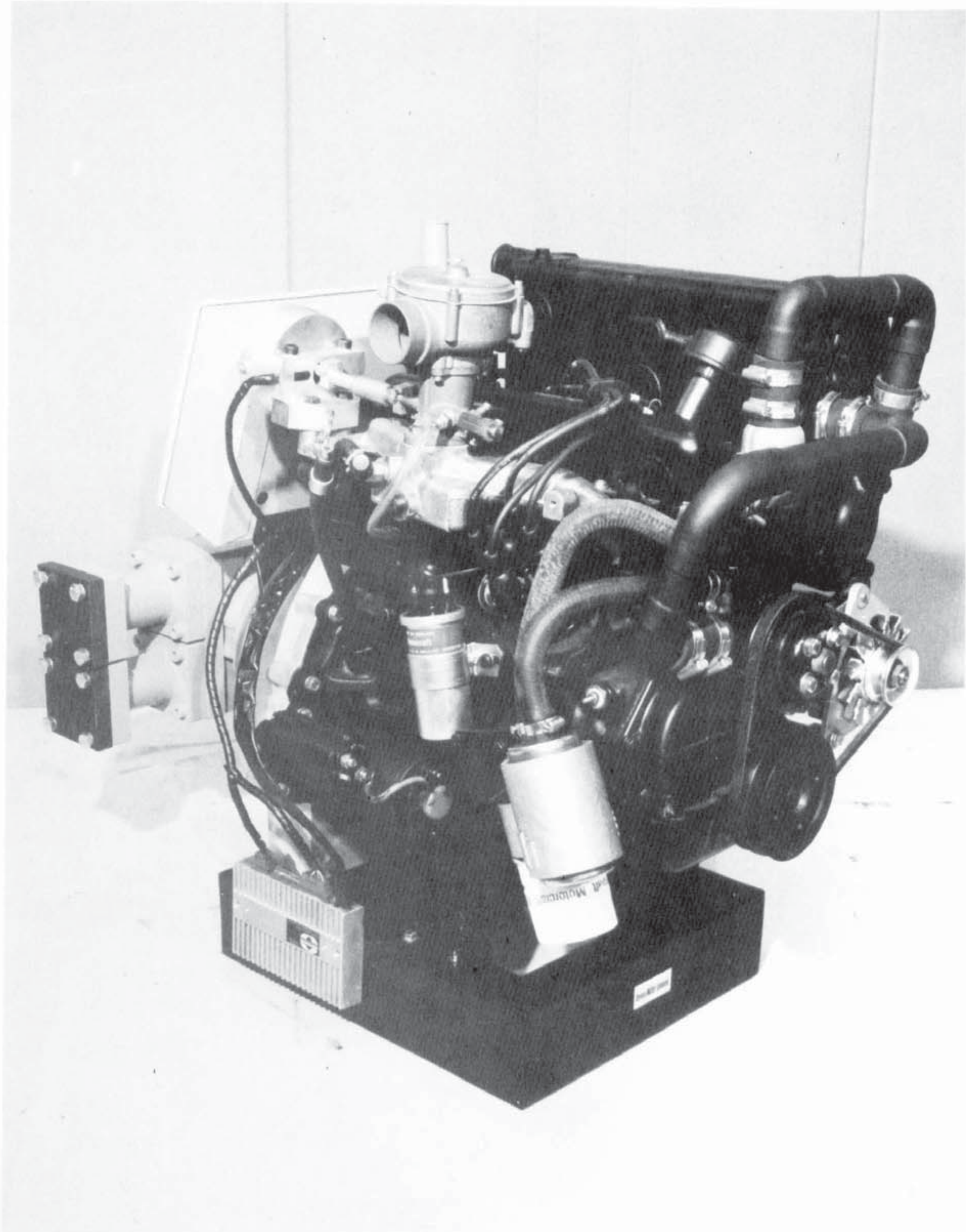

Plate A.1.1 Front Elevation of the Ford-Denco "Concept Engine Package"

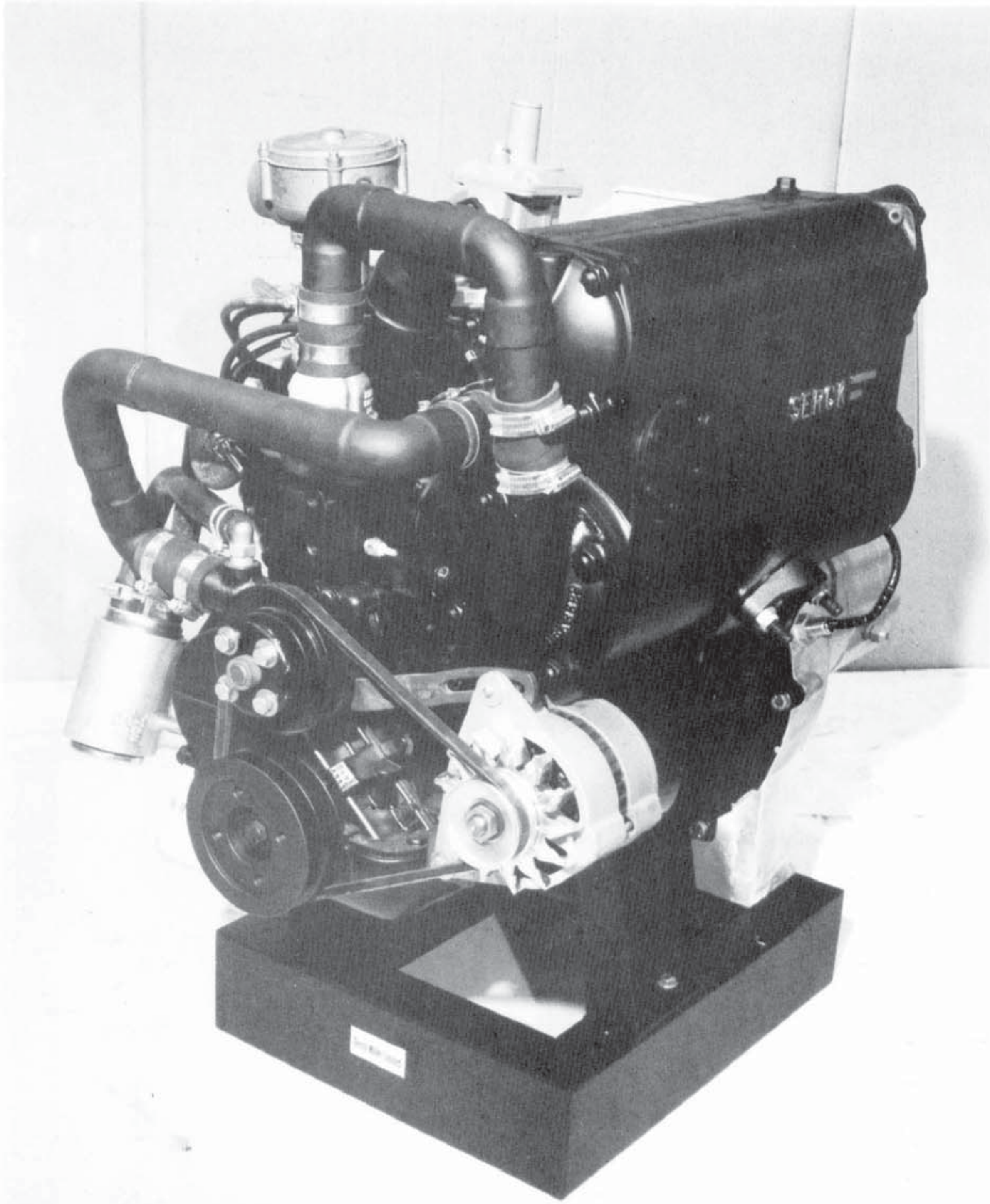

Plate A.1.2 Rear Elevation of the Ford-Denco "Concept Engine Package"

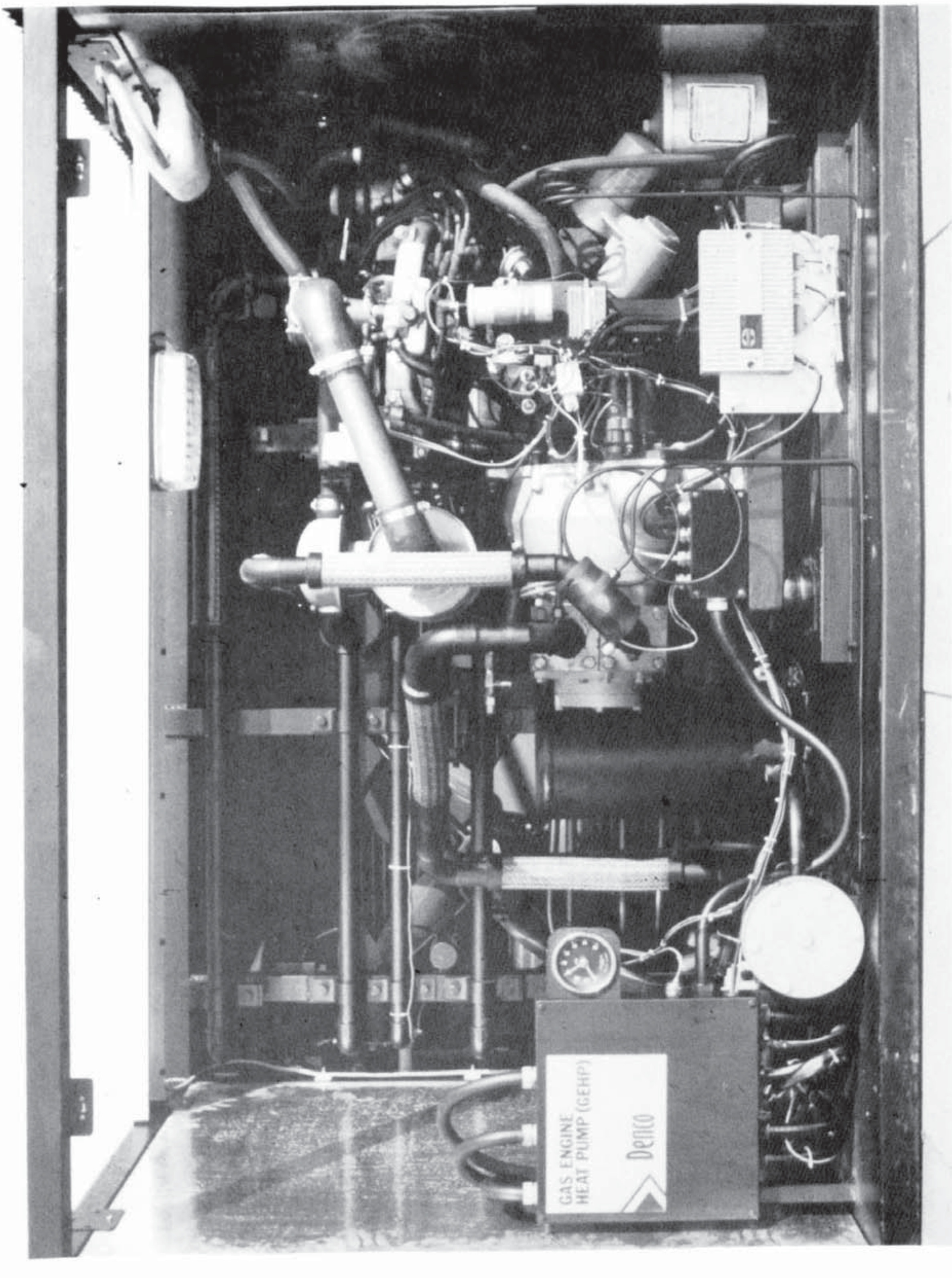

Plate A.1.3 Prototype Gas Engine Heat Pump (Engine-Compressor Module)

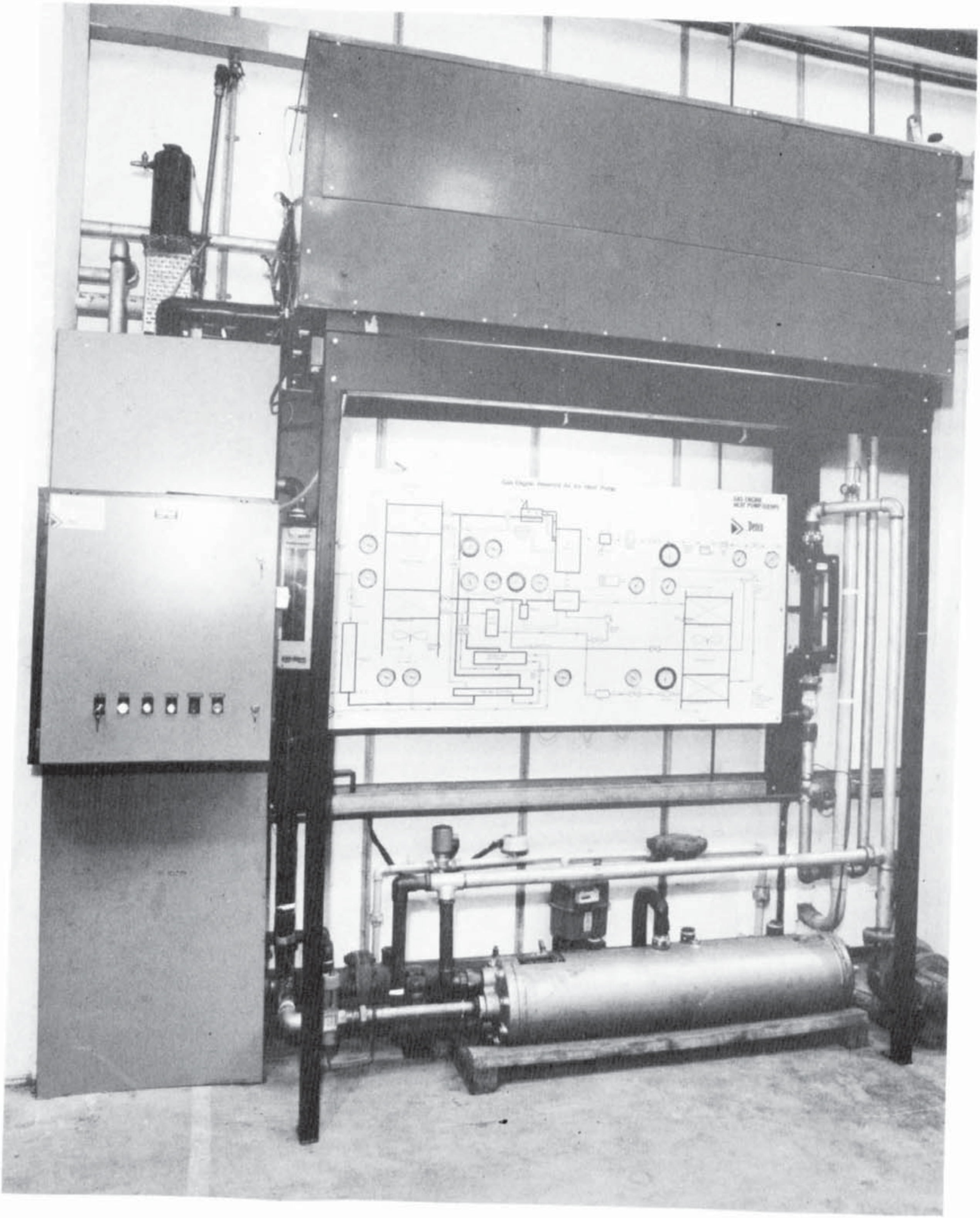

Plate A.1.4 Prototype Gas Engine Heat Pump (Ancillary Equipment)

Plate A.1.5 The Factory Area Heated Using The Prototype Gas Engine Heat Pump

Plate A.1.7 The Swimming Pool At Dean Close School, Cheltenham Heated Using A Gas Engine Heat Pump

APPENDIX 2

THEORETICAL ANALYSIS

APPENDIX 2.1

PROPOSED ECONOMIC ANALYSIS FOR GAS ENGINE HEAT PUMPS

Economic analysis of a gas engine driven heat pump is dependent upon the following factors:

a) Installed cost compared with conventional systems.

b) Life expectancy of the plant compared with conventional systems.

c) Annual rate of inflation : necessary to determine the accumulative fuel cost savings.

d) Annual interest rate : necessary to determine the present worth of the capital cost savings using a conventional system.

e) Annual fuel cost savings : this should include the electricity consumed by ancillary equipment.

f) Annual maintenance costs.

Let:

I = installed costs (£)
L = plant life expectancy (years)
A = annual running costs (£/year)
R = escalation rate and discount factor as defined by Manion [107]
Q = annual heat requirement (kWh)
F = fuel cost (£/kWh)
η = system efficiency
M = cost of maintenance contract annually (£/year)

$\overline{PER}$ = seasonal primary energy ratio, with a frosting allowance
QR = rated power consumption of system fans and pumps etc.

Q_H = rated heat output of heat pump at $\overline{PER}$
Q_B = rated heat output of boiler

Suffixes: H = heat pump; B = boiler; E = electricity

Derivation Of Annual Running Costs

$$\text{Annual running Costs} = \text{Primary Fuel Cost} + \text{Ancillary Fuel Cost} + \text{Maintenance Costs}$$

For the gas engine driven heat pump:

$$\text{Primary fuel Cost} = \frac{\text{Annual Heat Requirement x Fuel Cost}}{\text{Seasonal PER}}$$

Where:

$$\text{Seasonal PER} = \left[\frac{\Sigma \left(\text{PER at given temp.}\right) \times \left(\text{Annual Hrs at temp.}\right)}{\text{Total Hours}} \right] \times \text{Frosting Allowance}$$

$$\text{Ancillary Fuel Cost} = \text{Annual Operating Hours} \times \text{Rated Power For Fans etc.} \times \text{Elec. Cost}$$

Maintenance Costs = Cost Of Maintenance Contract

Thus:

$$A_H = Q\ (F_H/\ \overline{PER} + QR_H F_E/Q_H) + M_H$$

$$A_B = Q\ (F_B/\eta_B + QR_B F_E/Q_B) + M_B$$

Derivation of Installed Costs

If the life expectancy of the heat pump is less than the life expectancy of the boiler plant, the additional prime cost of the heat pump should be increased in the ratio:

$$\frac{\text{Life Expectancy Boiler Plant}}{\text{Life Expectance Heat Pump}} \quad \frac{(L_B)}{(L_H)}$$

Thus:

$$\textbf{Increase In Capital Cost For Heat Pump} = ((I_H - I_B)\ L_B/L_H)$$

Derivation Of Payback

$$\text{Payback} = \frac{\text{Increase In Capital Cost For Heat Pump}}{\text{Annual Fuel Cost Savings}}$$

$$\textbf{Payback} = \frac{((I_H - I_B)\ L_B/L_H)}{(A_B - A_H)}$$

Allowance For Inflation

The allowance for inflation in fuel costs and the interest lost on the additional capital has been derived by Manion [107].

Thus:

$$\text{Payback} = \left[\frac{((I_H - I_B)\ L_B/L_H)}{(A_B - A_H)}\right] R$$

APPENDIX 2.2

GAS ENGINE DRIVEN HEAT PUMP PERFORMANCE IN A SWIMMING POOL APPLICATION

a) Monthly performance:

Consider August 1983:

(See Appendix 3.10 for tabulated data)

$$\textbf{Rate of Fuel Usage} = \frac{\text{Volume Of Fuel x Calorific Value}}{\text{Operating Hours x 3600}} = \textbf{40.9 kW}$$

$$\textbf{Rate of Heat Generated} = \frac{\text{Total Heat Generated}}{\text{Operating Hours}} = \textbf{52.6 kW}$$

$$\textbf{Primary Energy Ratio} = \frac{\text{Heat Generated}}{\text{Fuel Used}} = \textbf{1.286}$$

b) Comparison of a gas engine driven heat pump with a gas boiler system with 70% efficiency for a swimming pool application:

(See Appendix 3.10 for tabulated data)

Fuel Used	=	1,828,420 ft^3
Heat Generated	=	625.6 MWh
Plant Operating Hours	=	7,263 hours
Rate of Heat Generation	=	86.14 kW
Rate of Fuel Usage	=	76.36 kW
Seasonal Primary Energy Ratio	=	1.13
Fuel Cost [106]	=	0.314 p/ft^3 (1.04 p/kWh)
Cost to Generate 625.6 MWh using a G.E.H.P. with a P.E.R. of 1.13	=	£5,757
Cost to Generate 625.6 MWh using a boiler with an efficiency of 70%	=	£9,294
Fuel Cost Saving	=	£3,537 (38.1%)

APPENDIX 3

TABLES OF RESULTS

APPENDIX 3.1

TABULATED RESULTS OF STRAIN GAUGE CALIBRATION FOR COMPRESSOR POWER ABSORBED ON THE PROTOTYPE GAS ENGINE DRIVEN HEAT PUMP

Applied Load (Nm)	Strain Gauge Output (mV)					
0	-0.02	-0.04	-0.04	-0.04	-0.04	-0.05
5.65	0.08	0.06	0.07	0.06	0.06	0.06
11.30	0.18	0.16	0.17	0.16	0.16	0.16
16.95	0.28	0.26	0.27	0.26	0.27	0.26
22.60	0.37	0.36	0.37	0.36	0.36	0.35
28.25	0.48	0.46	0.47	0.46	0.47	0.45
33.90	0.57	0.56	0.57	0.56	0.57	0.55
39.55	0.67	0.66	0.67	0.65	0.66	0.65
45.20	0.77	0.76	0.76	0.75	0.76	0.75
50.85	0.87	0.86	0.85	0.85	0.86	0.86
56.50	0.97	0.96	0.96	0.95	0.96	0.95
62.15	1.06	1.05	1.06	1.05	1.06	1.05
67.80	1.17	1.15	1.16	1.15	1.15	1.15
73.45	1.26	1.25	1.26	1.25	1.26	1.24
79.10	1.36	1.35	1.36	1.35	1.36	1.34
84.75	1.46	1.45	1.46	1.45	1.45	1.45
90.40	1.56	1.56	1.55	1.55	1.55	1.55

Table A3.1.1 Strain Gauge Calibration
Clockwise Rotation Of Compressor Shaft

Applied Load (Nm)	Strain Gauge Output (mV)					
0	0	0.01	0.01	0	0	0.01
5.65	0.09	0.09		0.09	0.09	0.09
11.30	0.19	0.18	0.19	0.18	0.19	0.19
16.95	0.29	0.28	0.29	0.28	0.29	0.28
22.60	0.39	0.38	0.39	0.38	0.39	0.38
28.25	0.48	0.48	0.49	0.48	0.49	0.48
33.90	0.58	0.58	0.58	0.58	0.59	0.58
39.55	0.68	0.67	0.68	0.67	0.68	0.68
45.20	0.78	0.77	0.78	0.78	0.78	0.78
50.85	0.88	0.87	0.88	0.87	0.88	0.87
56.50	0.97	0.97	0.98	0.97	0.98	0.97
62.15	1.07	1.07	1.07	1.07	1.07	1.07
67.80	1.17	1.17	1.17	1.17	1.17	1.17
73.45	1.27	1.26	1.27	1.27	1.27	1.26
79.10	1.37	1.36	1.37	1.36	1.37	1.36
84.75	1.47	1.46	1.47	1.46	1.47	
90.40	1.56	1.56	1.56	1.56	1.56	1.56

Table A3.1.2 Strain GaugeCalibration
Anticlockwise Rotation Of Compressor Shaft

APPENDIX 3.2

THERMOCOUPLE CALIBRATION

Thermocouple	Boiling Water (mV)	Boiling Water (°C)	Melting Ice (mV)	Melting Ice (°C)
Standard Thermometer	N.A.	99.8	N.A.	0.0
1 Natural Gas	4.08	100.4	+0.01	0.49
2 Exhaust	4.06	99.55	+0.01	0.49
3 Engine Coolant Flow	4.07	99.80	-0.02	-0.24
4 Engine Coolant Return	4.07	99.80	-0.01	0.01
5 Water Onto Serck	4.06	99.55	0.00	0.25
6 Air Off Evaporator	4.05	99.31	-0.02	-0.24
7 Air On Evaporator	4.08	100.04	0.01	0.49
8 Air Off Evaporator	4.07	99.80	-0.01	0.01
9 Air On Evaporator	4.05	99.31	-0.02	-0.24
10 Air Off Evaporator	4.07	99.80	0.00	0.25
11 Air On Evaporator	4.06	99.55	0.00	0.25
12 Refrig Onto Evapn (3)	12.21	99.80	-0.03	0.01
13 Refrig Off Evapn (3)	12.22	99.88	-0.06	-0.24
14 Ambient Air	4.07	99.80	0.01	0.49
15 Compr Suction (3)	12.20	99.71	-0.03	0.01
16 Compr Discharge (2)	8.15	99.92	0.01	0.37
17 Serck Water T (3)	12.21	99.80	-0.02	0.08
18 Refrig Onto Condr (2)	8.13	99.67	0.02	0.49
19 Refrig Off Condr (2)	8.15	99.92	-0.03	-0.12
20 Condr Water T (3)	12.20	99.71	-0.02	0.08
21 Water Onto Condr	4.07	99.80	0.00	0.25
22 Internal Temperature	4.05	99.31	0.01	0.49

Table A3.2.1 Thermocouple Calibration For The Prototype Gas Engine Heat Pump

Thermocouple	Boiling Water (°C)	Melting Ice (°C)
Standard Thermometer	99.7	0.0
1 Water Temp Off Condr	99.5	-0.1
2 Water Temp On Condr	99.5	-0.3
3 Compr Suction Temp	99.7	0.0
4 Compr Discharge Temp	99.7	0.1
5 Refrig Temp Onto Condr	99.9	0.1
6 Refrig Temp Off Condr	99.6	0.0
7 Refrig Temp Onto Evapr	99.8	-0.1
8 Refrig Temp Off Evapr	99.8	0.0
9 Air Temp Off Evaporator	99.7	0.2
10 Air Temp Onto Evaporator	99.7	0.1

Table A3.2.2 Thermocouple Calibration For The Fluidised Bed Reactor

APPENDIX 3.3

PRESSURE TRANSDUCER/GAUGE CALIBRATIONS

Transducer	Transducer Output at 105.5 kN/m^2 (mV)	Transducer Output Corrected to 1 bar (mV)	Absolute Zero Correction Factor (mV)
1 Pressure onto Condenser	0.895	0.84	+0.16
2 Pressure off Condenser	1.005	0.95	+0.05
3 Compressor Suction Pressure	0.775	0.72	+0.28
4 Compressor Discharge Pressure	0.985	0.93	+0.07
5 Pressure Onto Evapr	1.025	0.97	+0.03
6 Pressure Off Evapr	1.265	1.21	-0.21

Table A3.3.1 Pressure Transducer Zero Offset Condition For The Prototype Gas Engine Heat Pump Barometric Pressure: 105.5 kN/m^2

Gauge Pressure For N.P.L. Pressure Gauge (bar)	Gauge Pressure Measured With Bourdon Tube Gauges (bar)				
	(1) Compr. Suction Pressure	(2) Pressure Onto Condr.	(3) Pressure Off Condr.	(4) Compr. Suction Pressure	(5) Pressure Off Evapr.
0	0	0	0	0	0
1	1.07	0.79	0.93	-	-
3	2.93	2.86	2.90	2.75	3.21
5	4.93	4.86	4.79	4.79	5.14
7	6.93	7.28	6.71	7.00	7.21
9	8.84	9.48	8.62	9.00	9.28
11	-	-	-	11.00	11.14
13	-	-	-	13.14	13.14

Table A3.3.2 Bourdon Tube Pressure Gauge Calibration For The Fluidised Bed Reactor

APPENDIX 3.4

PROTOTYPE GAS ENGINE DRIVEN HEAT PUMP STEADY STATE RESULTS

The results contained in this Appendix are presented graphically. A full set of data is presented for the heat rejected at the condenser, and sample data at random speeds are presented for all other variables.

The curves superimposed on this data are not "best fit" curves, but are the curves generated by the empirical relationships for the total data obtained. The empirical relationships are contained at the end of this Appendix.

The symbols used throughout this Appendix are defined as:

Ambient relative humidity range below 60%	○
60 to 69%	□
70 to 79%	●
80 to 89%	▲
90 to 100%	■

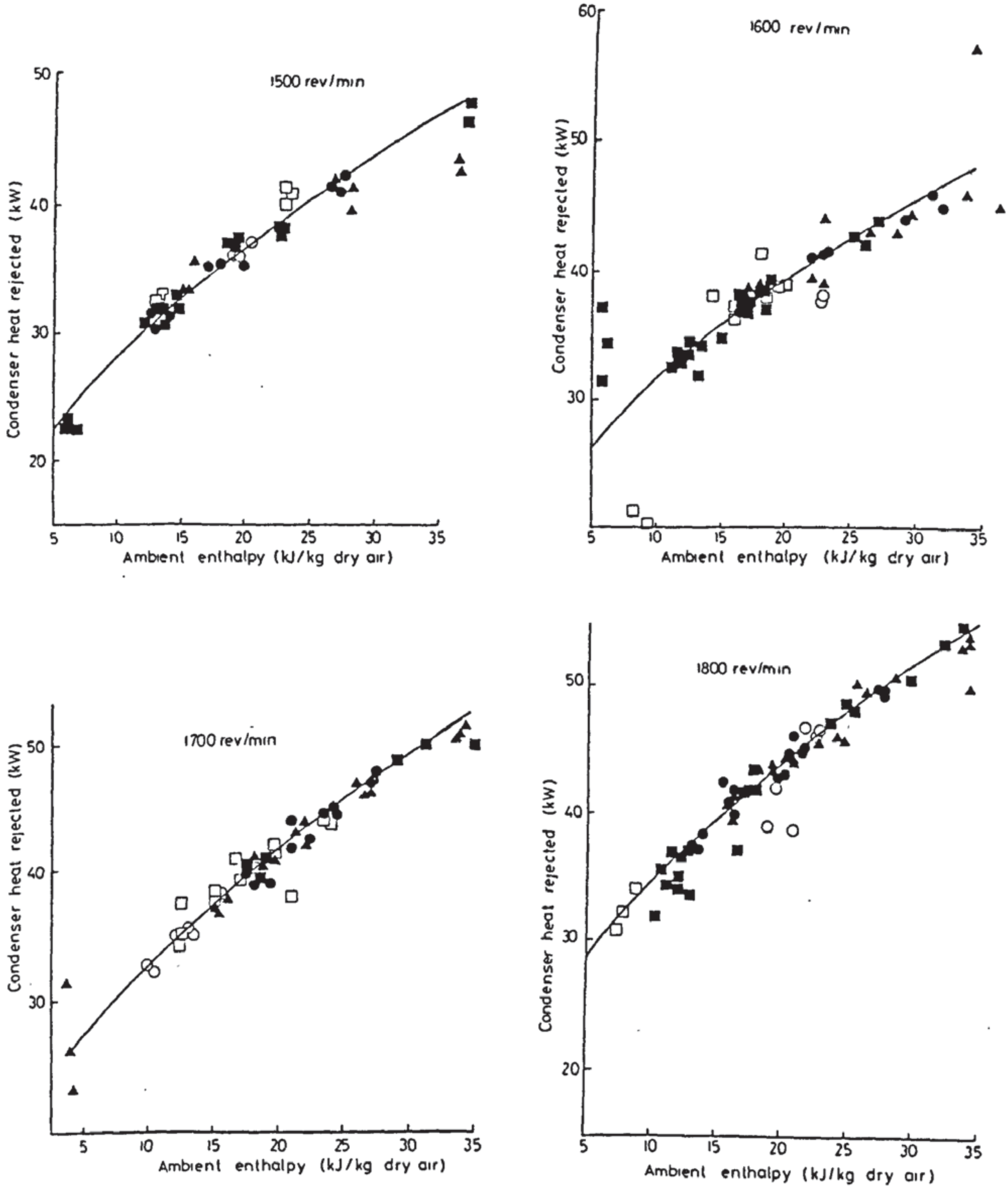

Figure A.3.4.1 Gas Engine Heat Pump Performance Data
Condenser Heat Rejected v Ambient Enthalpy
Engine Speed 1500 Rev/Min - 1800 Rev/Min

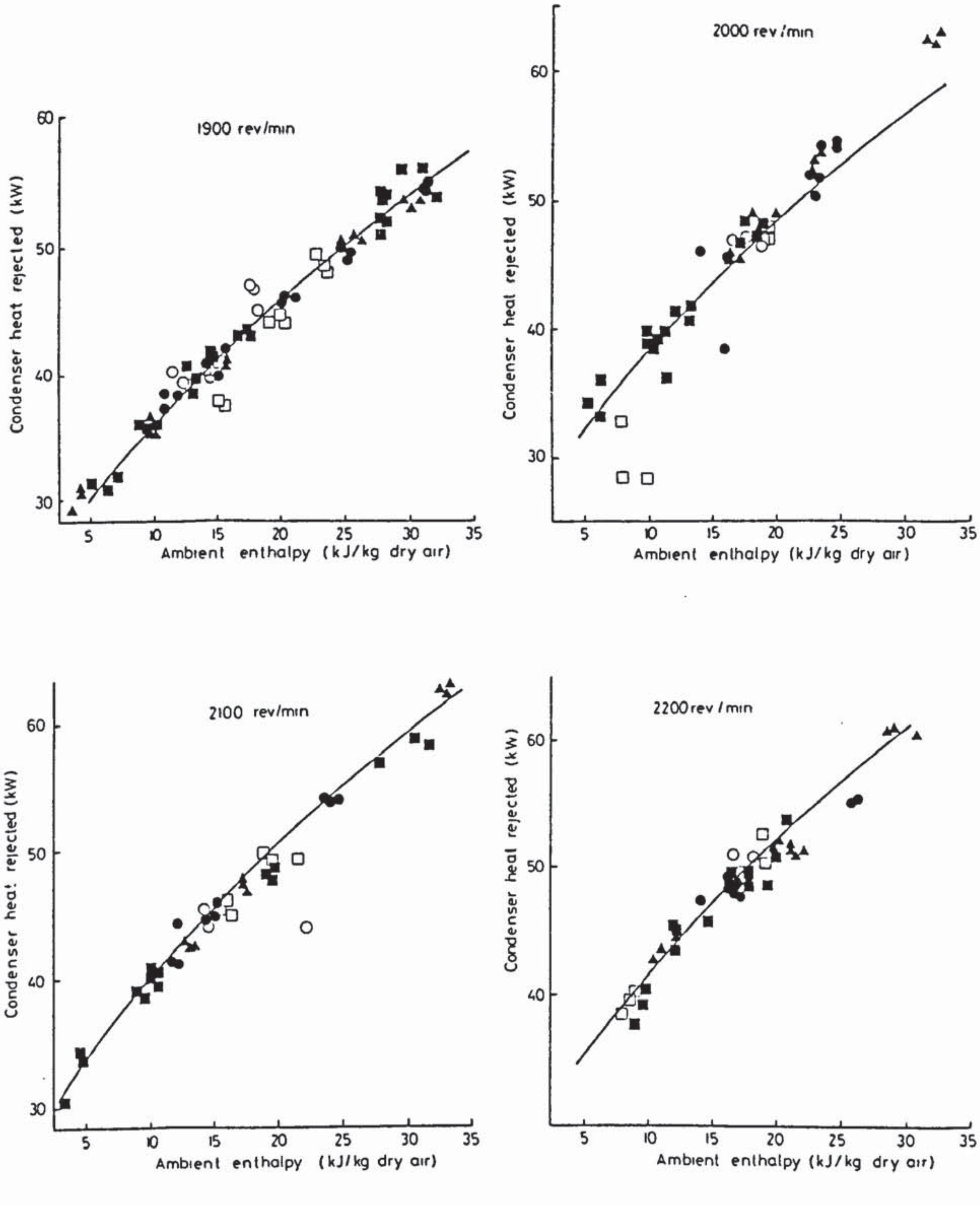

Figure A.3.4.2 Gas Engine Heat Pump Performance Data Condenser Heat Rejected v Ambient Enthalpy Engine Speed 1900 Rev/Min - 2200 Rev/Min

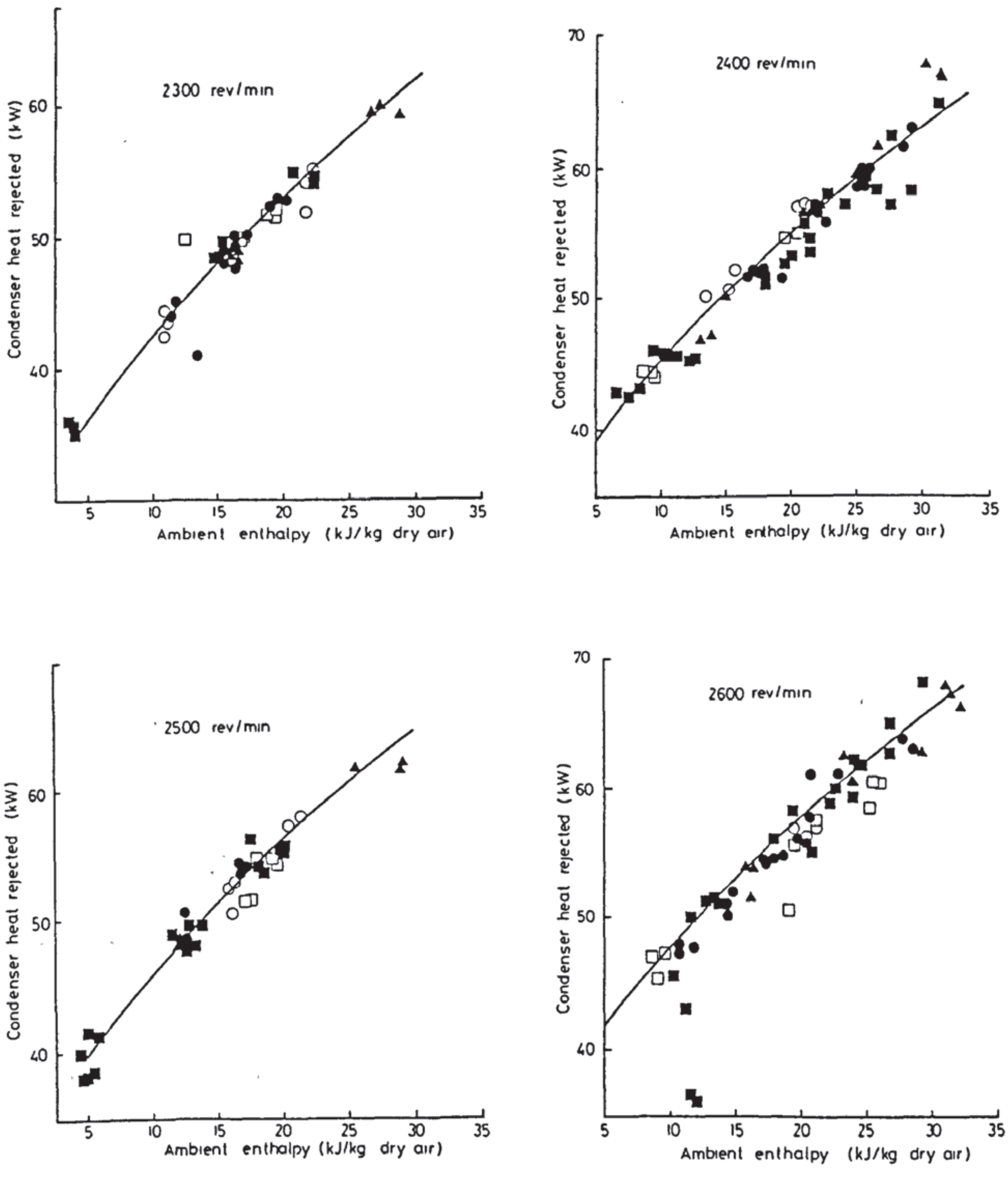

Figure A.3.4.3 Gas Engine Heat Pump Performance Data Condenser Heat Rejected v Ambient Enthalpy Engine Speed 2300 Rev/Min - 2600 Rev/Min

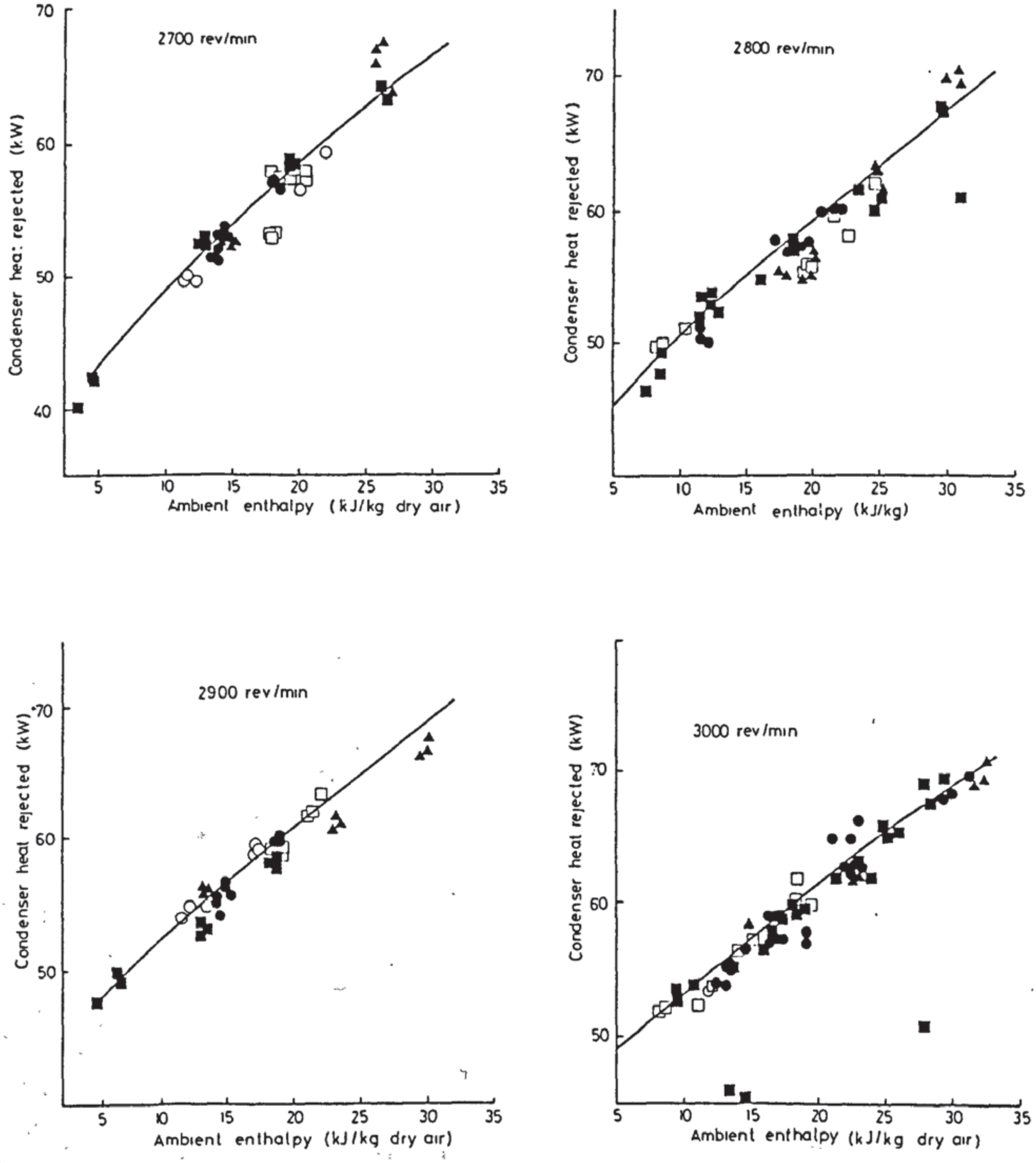

Figure A.3.4.4 Gas Engine Heat Pump Performance Data
Condenser Heat Rejected v Ambient Enthalpy
Engine Speed 2700 Rev/Min - 3000 Rev/Min

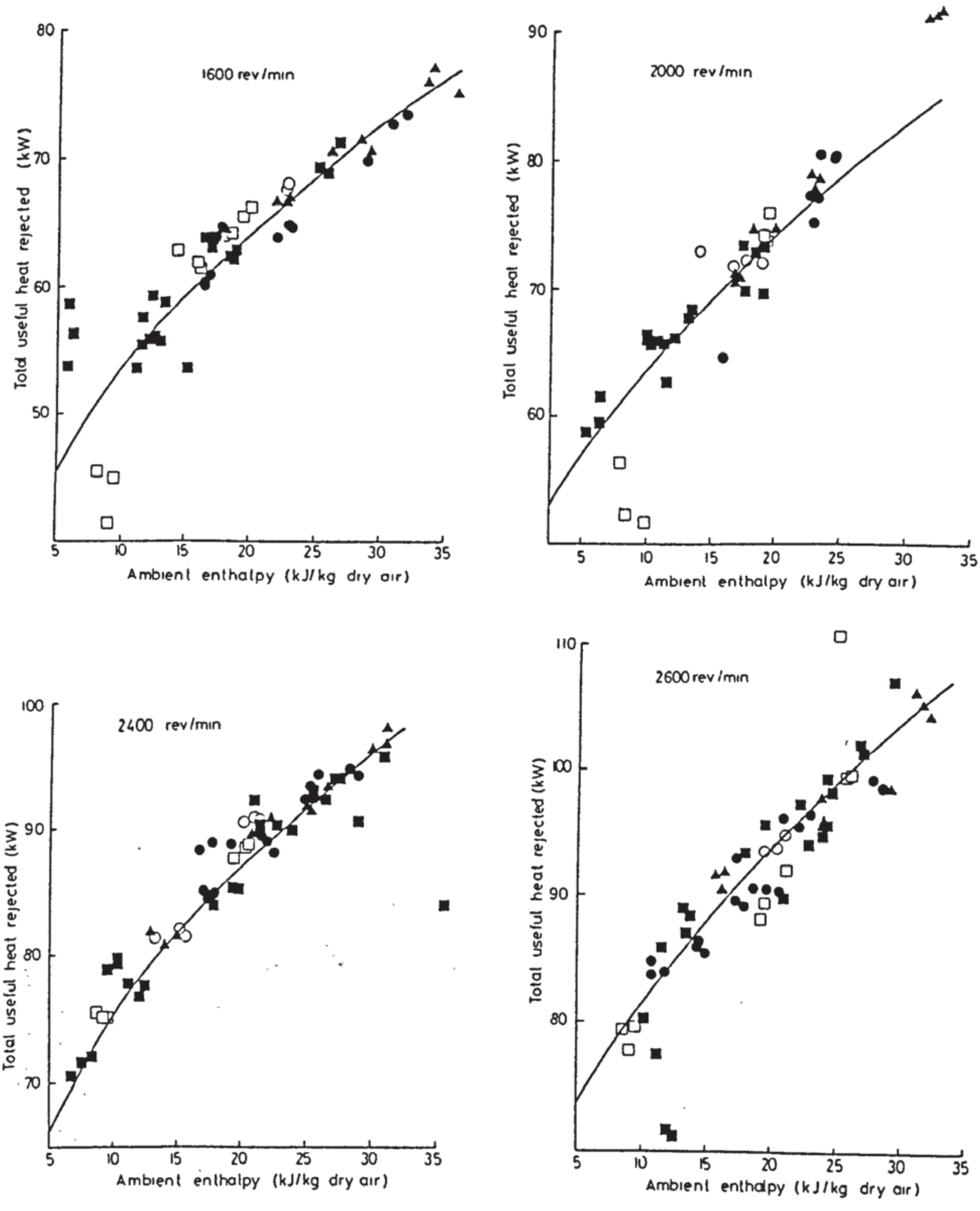

Figure A.3.4.5 Gas Engine Heat Pump Performance Data
Total Useful Heat Rejected v Ambient Enthalpy

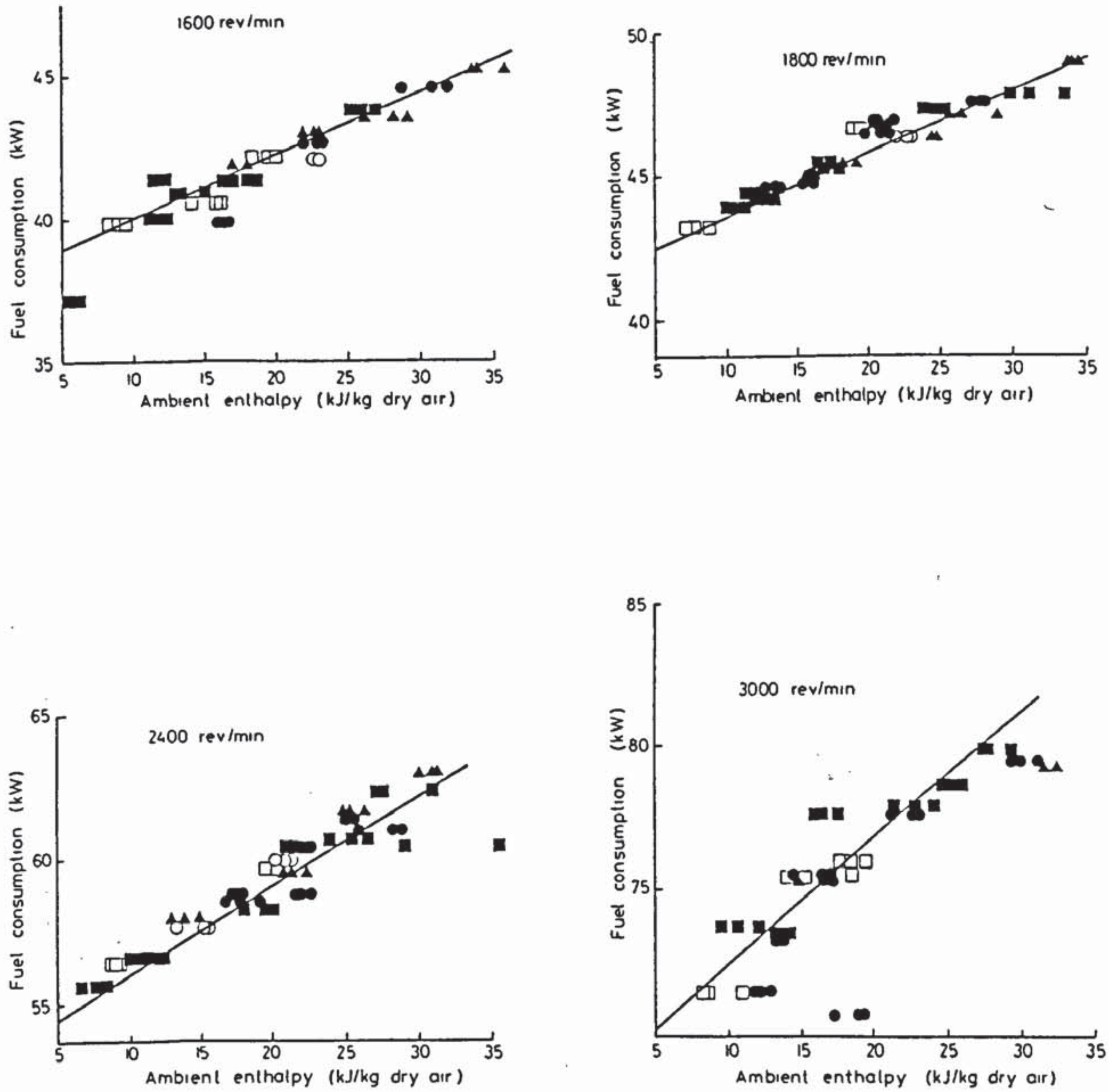

Figure A.3.4.6 Gas Engine Heat Pump Performance Data
Fuel Consumption v Ambient Enthalpy

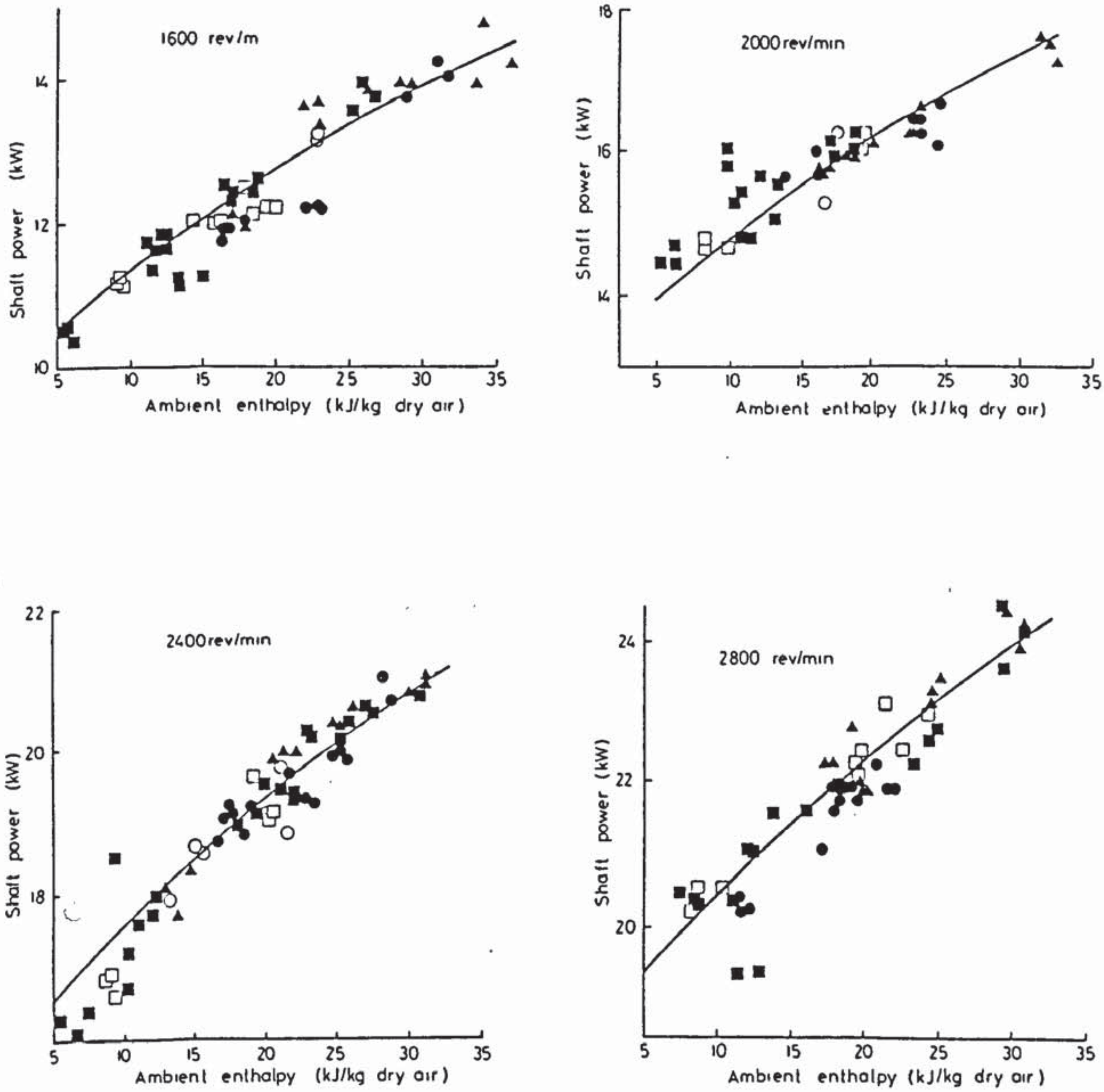

Figure A.3.4.7 Gas Engine Heat Pump Performance Data
Shaft Power v Ambient Enthalpy

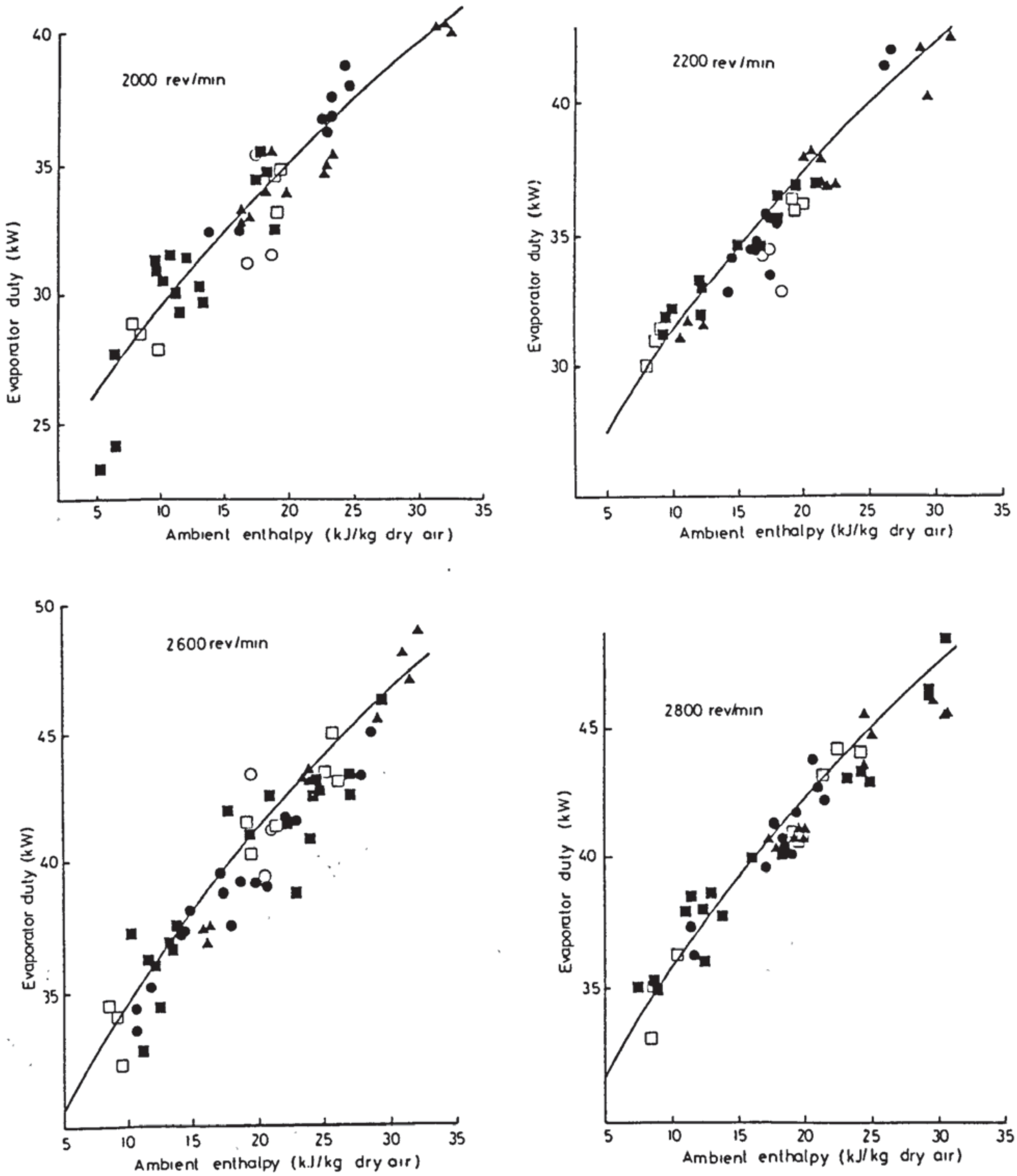

Figure A.3.4.8 Gas Engine Heat Pump Performance Data
Evaporator Duty v Ambient Enthalpy

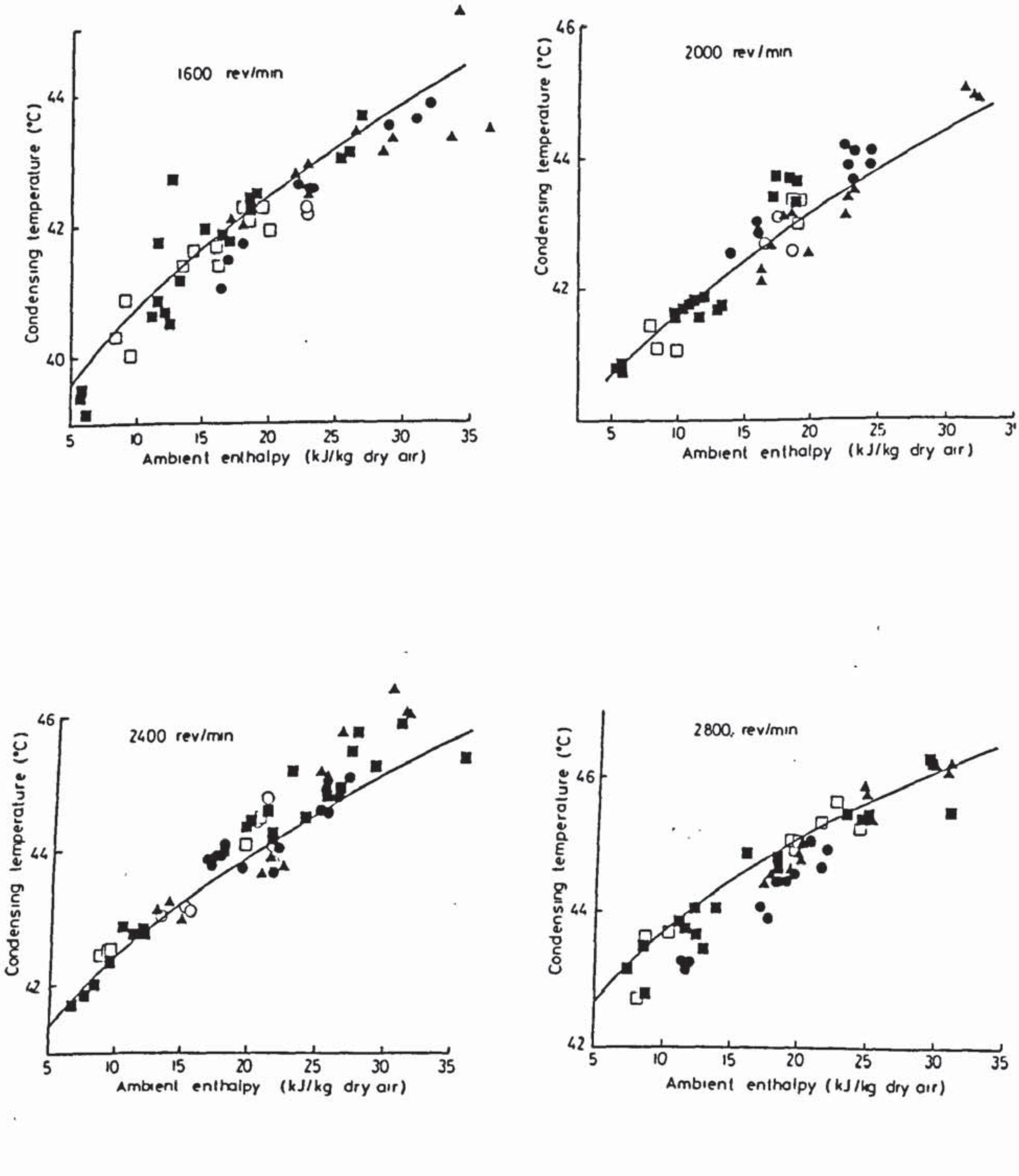

Figure A.3.4.9 Gas Engine Heat Pump Performance Data
Condensing Temperature v Ambient Enthalpy

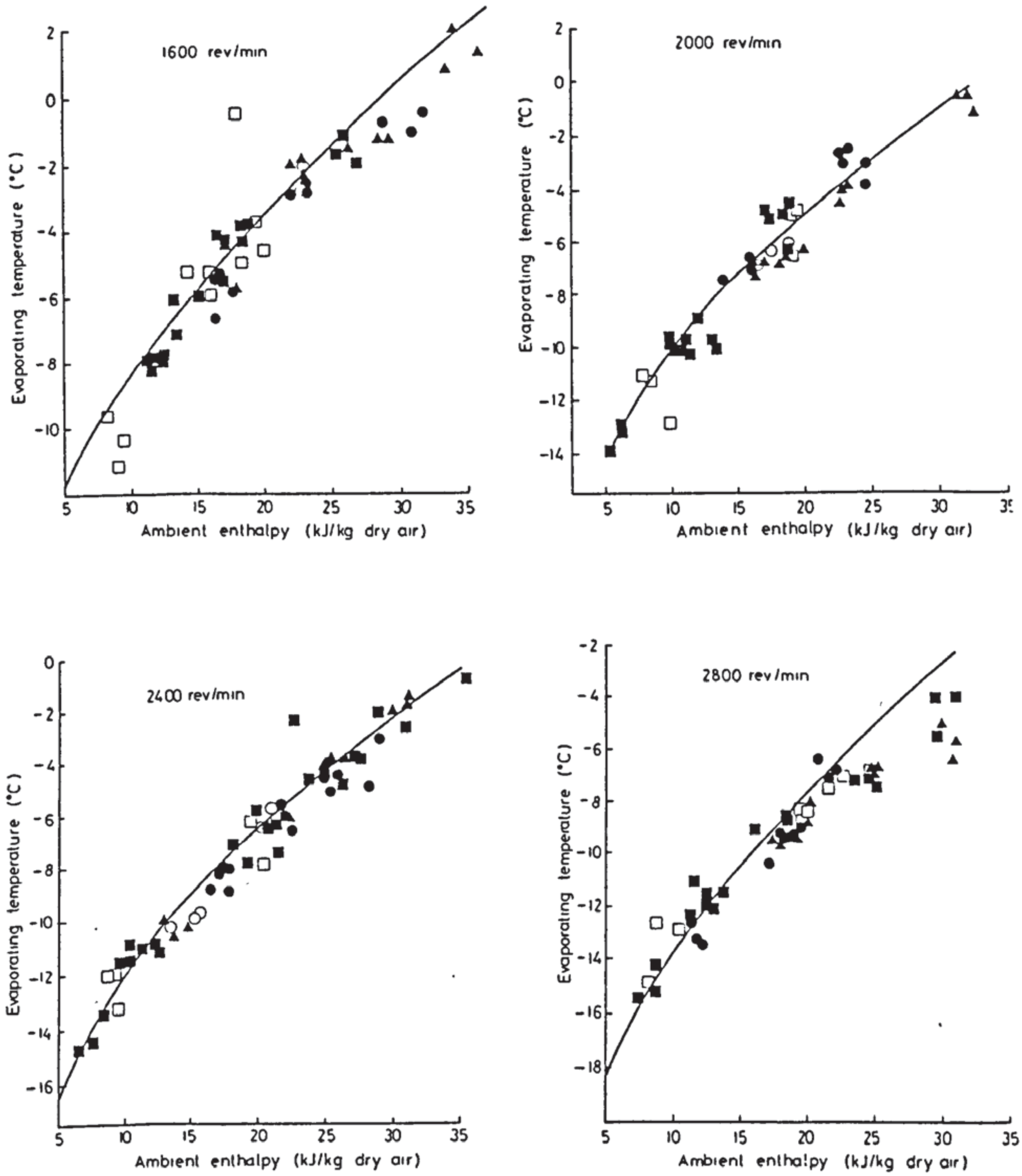

Figure A.3.4.10 Gas Engine Heat Pump Performance Data
Evaporating Temperature v Ambient Enthalpy

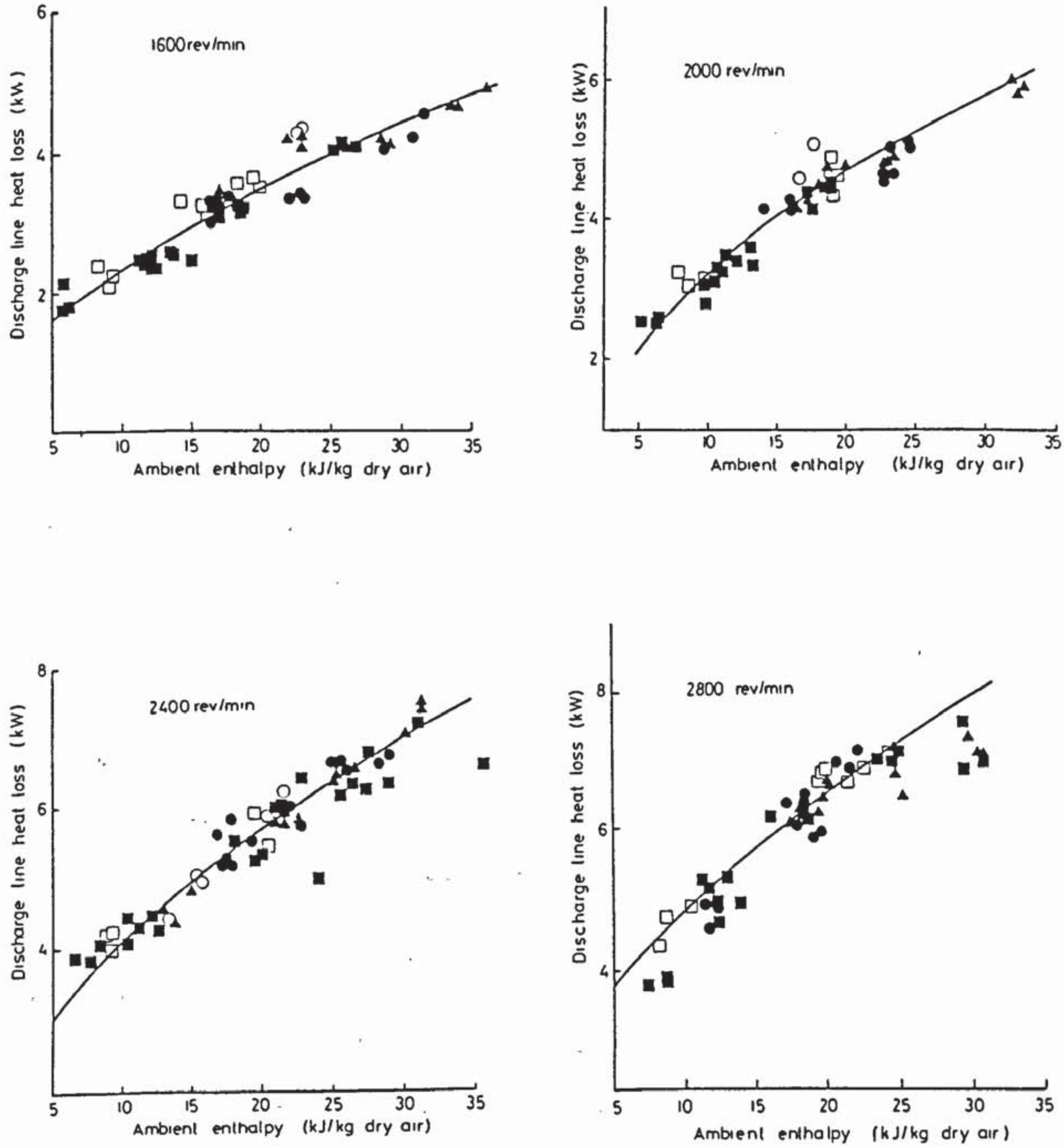

**Figure A.3.4.11 Gas Engine Heat Pump Performance Data
Discharge Line Heat Loss v Ambient Enthalpy**

Mathematical Models Derived From The Steady State Performance Data For The Prototype Gas Engine Heat Pump

1. Total System Output (Q_T):

$$Q_T = 12.4155+5.8831\times10^{-6}N^2+(6.3892+1.0717\times10^{-3}N)H^{1/2}$$

2. Condenser Output (Q_C):

$$Q_C = 1.1007\times10^{-3}N^{1.327}+H(-0.7317+1.923\times10^{-3}N-4.7417\times10^{-7}N^2)+H^2(2.4172\times10^{-3}-1.3699\times10^{-5}N+4.309\times10^{-9}N^2)$$

3. Fuel Consumption (Q_F):

$$Q_F = 29.6824+5.3794\times10^{-6}N^2+H(0.2679-1.1572\times10^{-4}N+6.4055\times10^{-8}N^2)$$

4. Shaft Power (W):

$$W = 1/(-0.0896+7.5941/N^{0.5})+H(-0.97+1.0764\times10^{-3}N-2.3583\times10^{-7}N^2)+H^2(0.0253-2.5594\times10^{-5}N+5.8384\times10^{-9}N^2)$$

5. Evaporator Performance (Q_E):

$$Q_E = 40.626-36305.3/N+H(0.9898-997762/N^2) + H^2(-0.025+1.75\times10^{-5}N-3.964\times10^{-9}N^2)$$

6. Discharge Line Heat Loss (Q_L):

$$Q_L = 2.5553-3.5314\times10^{-3}N+1.046\times10^{-6}N^2+H^{0.5}(-1.48+2.1087\times10^{-3}N-4.0295\times10^{-7}N^2)$$

7. Condensing Temperature (T_c):

$$T_c = 34.5783+7.7472\times10^{-7}N^2+H^{0.5}(1.8215-3.0509\times10^{-4}N)$$

8. Evaporating Temperature (T_e):

$$T_e = 1/(0.011-2.415/N^{0.5})+N^{0.5}(3.041+2.3924\times10^{-4}N+1.2639\times10^{-7}N^2)$$

9. Coefficient of Performance (C.O.P.):

$$C.O.P = \frac{Q_c}{W}$$

10. Primary Energy Ratio (P.E.R.)

$$P.E.R. = \frac{Q_T}{Q_F}$$

Where:
N is the engine speed in rev/min
H is the ambient enthalpy in kJ/kg of dry air.

APPENDIX 3.5
PROTOTYPE GAS ENGINE DRIVEN HEAT PUMP TRANSIENT RESULTS

TIME (Min)	% STEADY STATE CONDENSER OUTPUT (Ambient Enthalpy (kJ/kg dry air)								
	(5)			(10)			(15)		
	Test 1	Test 2	Test 3	Test 1	Test 2	Test 3	Test 1	Test 2	Test 3
0	100	100	100	100	100	100	100	100	100
5	100.8	98.1	96.9	104.6	100	99.1	100.1	101.2	102.2
10	100	96.2	94.4	102.2	99.6	98.0	100.4	102.2	103.4
15	97.8	94.8	92.9	96.1	95.2	94.4	99.5	101.3	103.1
20	94.5	89.6	88.0	93.2	91.8	89.4	99.2	100.6	101.4
25	87.6	85.1	83.7	92.4	90.7	90.1	98.7	99.6	100.2
30	80.2	79.9	79.3	89.0	86.6	86.2	97.8	98.2	99.1
35	74.1	72.5	71.6	87.1	84.7	84.3	96.5	97.2	98.4
40	63.2	64.3	64.4	84.2	80.1	79.5	95.8	96.4	97.4
45	58.0	57.0	56.4	80.0	75.1	72.2	93.5	95.3	96.7
50	41.5	47.4	41.2	71.4	69.2	68.4	90.8	92.5	95.0
55	41.6	44.2	38.1	64.2	63.7	59.9	88.0	89.4	90.8
60	31.6	31.5	31.4	60.0	58.2	56.0	85.4	88.1	86.7
65				51.8	52.4	48.7	83.2	85.9	84.8
70				46.1	48.3	47.2	81.0	82.6	82.4
75							77.8	79.1	80.6
80							75.1	73.7	76.2
85							70.2	67.4	68.8
90							65.7	62.2	59.6

Table A3.5.1 Condenser Heat Rejected During Transient Experiments Engine Speed 1800 Rev/Min

TIME (Min)	% STEADY STATE CONDENSER OUTPUT (Ambient Enthalpy (kJ/kg dry air)							
	(5)		(10)		(15)		(20)	
	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2
0	100	100	100	100	100	100		
5	95.4	99.1	99.5	100	100.2	101.4		
10	93.8	96.4	98.6	99.4	100.4	101.0		
15	90.6	91.6	96.7	97.4	100.1	100.6		
20	86.1	85.3	95.1	95.8	99.4	98.4		
25	80.3	77.4	91.8	92.0	98.2	96.2		
30	70.8	68.2	85.0	83.7	97.7	95.3	100	99.5
35	58.0	55.8	78.0	76.8	95.1	94.0	98.4	97.8
40	46.1	43.6	70.4	72.0	91.5	92.1	97.6	95.6
45			61.9	63.4	87.4	89.5	96.1	93.6
50			52.8	54.1	82.0	84.2	94.0	91.0
55			42.0	41.4	76.4	78.1	91.8	88.5
60					70.8	73.5	88.4	85.8
65					64.1	66.4	85.3	83.0
70					55.7	61.0	80.9	79.8
75					45.2	55.8	74.3	75.1
80					34.6	46.3	66.1	68.7
85							56.2	61.2
90							47.9	51.8
95							38.8	40.1

Table A3.5.2 Condenser Heat Rejected During Transient Experiments Engine Speed 2200 Rev/Min

TIME (Min)	% STEADY STATE CONDENSER OUTPUT (Ambient Enthalpy (kJ/kg dry air)							
	(5)		(10)		(15)		(20)	
	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2
0	100	100	100	100	100	100	100	100
5	100.3	97.9	102.8	99.4	99.4	101.3		
10	96.1	92.4	100.4	98.2	99.6	100.7		
15	86.5	86.4	98.1	96.4	98.4	100.4		
20	76.7	79.2	95.3	92.9	97.8	99.0	100	100
25	67.0	66.8	90.7	87.9	93.7	94.6	100.4	99.6
30	54.4	48.3	84.4	81.3	90.0	90.1	99.5	98.1
35	37.2	32.2	76.2	73.0	87.9	85.9	97.9	97.3
40			67.0	62.4	82.4	81.6	94.3	93.2
45			54.9	50.0	73.7	76.3	90.6	88.7
50			36.8	38.5	67.8	70.8	86.0	87.7
55					61.5	66.5	80.5	85.3
60					53.0	59.0	76.8	79.6
65					42.9	45.3	73.1	73.8
70					37.6	34.7	62.9	65.9
75							56.7	57.4
80							50.4	53.8

Table A3.5.3 Condenser Heat Rejected During Transient Experiments Engine Speed 2600 Rev/Min

TIME (Min)	% STEADY STATE TEMPERATURE APPROACH (Ambient Enthalpy (kJ/kg dry air)								
	(5)			(10)			(15)		
	Test 1	Test 2	Test 3	Test 1	Test 2	Test 3	Test 1	Test 2	Test 3
0	100	100	100	100	100	100	100	100	100
5	95.8	97.1	96.2	100.8	99.9	100.1	100.7	101.2	102.9
10	93.4	93.1	93.2	100.2	100.7	99.8	101.3	100.8	101.7
15	88.5	89.2	87.1	98.0	98.4	98.3	99.3	99.8	99.0
20	84.0	83.6	84.5	98.4	98.5	98.2	98.3	99.4	98.1
25	77.4	79.2	78.7	96.4	94.7	94.0	99.5	98.5	94.2
30	77.6	76.8	74.2	91.4	92.6	91.9	97.1	97.0	91.9
35	71.3	75.7	71.9	85.6	88.1	88.3	95.4	95.2	95.7
40	65.7	63.9	65.7	83.6	85.6	83.6	93.9	92.7	91.9
45	58.9	57.0	60.2	76.3	78.7	77.5	88.8	89.6	88.7
50	53.1	51.9	53.7	69.7	71.4	70.0	83.3	84.0	85.0
55	44.8	44.0	42.9	64.5	62.9	63.1	76.4	78.1	78.2
60	39.0	41.7	40.4	57.0	56.2	55.8	70.9	71.9	72.5
65				44.0	48.8	44.3	63.2	66.0	64.2
70				39.2	36.7	37.1	59.3	58.2	60.1
75							53.9	51.7	53.4
80							49.2	46.2	44.5

Table A3.5.4 Evaporating Temperature During Transient Experiments Engine Speed 1800 Rev/Min

TIME (Min)	% STEADY STATE TEMPERATURE APPROACH (Ambient Enthalpy (kJ/kg dry air)							
	(5)		(10)		(15)		(20)	
	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2
0	100	100	100	100	100	100		
5	98.7	97.3	97.1	93.8	99.4	98.2		
10	94.6	96.2	93.1	90.9	95.4	96.7		
15	92.1	91.4	88.2	89.3	94.3	95.9	100	100
20	82.1	84.0	87.2	86.0	92.5	96.2	99.0	98.6
25	76.4	75.9	80.4	81.9	90.4	91.8	98.1	97.6
30	67.1	65.3	73.9	78.4	86.4	87.1	96.0	95.0
35	58.8	56.1	72.5	73.0	89.9	85.4	94.7	95.1
40	42.5	45.0	68.4	65.4	82.1	81.4	93.2	91.6
45			60.6	59.0	78.2	79.7	90.3	87.7
50			56.4	51.2	70.8	74.1	87.9	85.1
55			47.2	40.8	69.9	65.7	81.4	80.3
60					66.2	64.9	76.1	75.4
65					60.4	58.1	71.2	70.3
70					50.1	55.0	65.0	63.7
75					46.3	49.0	59.6	61.5
80					42.9	39.8	53.5	51.3
85							46.5	44.3
90							39.1	42.5

Table A3.5.5 Evaporating Temperature During The Transient Experiments Engine Speed 2200 Rev/Min

TIME (Min)	% STEADY STATE TEMPERATURE APPROACH (Ambient Enthalpy (kJ/kg dry air)							
	(5)		(10)		(15)		(20)	
	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2
0	100	100	100	100	100	100	100	100
5	96.1	96.6	98.1	99.3	99.0	98.3	102.5	99.7
10	89.3	91.6	92.6	94.7	97.3	96.2	101.3	98.4
15	87.2	84.1	88.2	90.5	94.4	93.5	99.7	97.9
20	78.9	74.3	82.5	83.8	89.8	90.9	98.4	97.2
25	66.1	69.9	75.2	78.5	86.3	86.1	95.8	94.0
30	57.8	60.3	70.5	72.4	79.2	80.9	92.7	91.0
35	49.4	53.2	63.4	62.4	74.1	74.8	88.4	85.5
40			55.2	58.7	68.5	68.4	81.5	79.9
45			49.5	51.2	61.9	63.0	73.2	74.0
50			42.4	44.9	54.7	52.9	65.3	67.4
55					45.6	42.3	58.1	58.6
60					37.4	39.7	49.1	51.0
65					29.7	26.5	42.4	44.8
70					22.4	23.9	32.5	35.6
75							24.3	27.7
80							19.5	20.9

Table A3.5.6 Evaporating Temperature During Transient Experiments Engine Speed 2600 Rev/Min

TIME (Min)	% STEADY STATE POWER ABSORBED (Ambient Enthalpy (kJ/kg dry air)								
	(5)			(10)			(15)		
	Test 1	Test 2	Test 3	Test 1	Test 2	Test 3	Test 1	Test 2	Test 3
0	100	100	100	100	100	100	100	100	100
5	100.5	101.6	102.0	102.4	100.1	101.1	101.8	101.8	100
10	99.3	100	103.3	103.6	100.1	101.1	102.4	101.7	100
15	100.5	98.3	103.2	101.7	99.4	101.4	102.5	100.8	101.5
20	100	97.4	103.2	99.2	98.5	101.4	102.8	101.7	101.5
25	100	98.3	103.6	100	97.5	101	102.6	101.7	97.6
30	99.5	101.6	103.5	100	92.5	100.6	102.8	101.7	100.7
35	99.1	98.3	102.7	100.8	94.7	100.4	102.9	101.7	98.4
40	98.7	94.9	103.2	100	96.7	99.7	103.1	101.7	95.2
45	93.6	92.4	103.3	101.7	98.3	99.1	101.8	101.7	92.8
50	94.9	94.9	102.2	99.2	94.7	98.1	102.9	101.7	95.2
55	95.5	89.9	98.8	100.8	94.5	98.1	102.3	100.8	94.4
60	92.7	94.9	97.4	98.3	94.1	97.6	101.3	100.8	87.3
65				97.5	95.9	92.9	99.9	99.1	86.5
70				92.5	94.7	93.3	102.1	98.2	88.9
75							99.7	95.7	88.9
80							100.6	96.5	86.5
85							100.1	96.5	86.5
90							99.1	98.2	89.6

Table A3.5.7 Compressor Power Absorbed During Transient Experiments Engine Speed 1800 Rev/Min

TIME (Min)	% STEADY STATE POWER ABSORBED (Ambient Enthalpy (kJ/kg dry air)							
	(5)		(10)		(15)		(20)	
	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2
0	100	100	100	100	100	100	100	100
5	96.1	101.7	100.8	98.9	104.4	100	101.6	101.6
10	94.2	100.9	100.8	98.3	102.7	100	102.4	102.0
15	93.6	99.1	98.5	97.8	101.7	100	102.4	102.0
20	92.2	95.6	98.5	96.0	100.8	99.6	104.9	101.6
25	91.8	96.5	98.5	96.1	99.1	99.4	104.9	100.8
30	92.1	96.5	97.7	95.7	99.1	99.1	104.0	100
35	89.9	94.6	97.7	94.1	100.0	98.2	103.2	100
40			92.3	94.0	95.6	95.6	102.4	100
45			89.3	95.2	94.7	94.2	100.8	100
50			94.7	95.5	96.4	93.2	100	100
55					96.4	94.2	95.9	100
60					95.5	93.2	100.8	99.1
65					94.6	92.0	96.7	98.4
70					94.6	91.7	96.7	97.6
75							90.1	96.1
80							91.8	95.5

Table A3.5.8 Compressor Power Absorbed During Transient Experiments Engine Speed 2200 Rev/Min

TIME (Min)	% STEADY STATE POWER ABSORBED (Ambient Enthalpy (kJ/kg dry air)							
	(5)		(10)		(15)		(20)	
	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2
0	100	100	100	100	100	100	100	100
5	99.1	100.8	101.5	99.4	100	89.6	101.7	103.4
10	99.0	98.3	98.4	98.7	99.4	89.2	102.5	101.5
15	99.9	101.6	100.7	98.7	99.9	89.8	101.1	102.5
20	99.5	96.6	99.2	96.4	99.9	87.7	101.1	102.1
25	98.8	96.6	98.5	97.3	98.1	87.3	103	101.8
30	96.4	94.9	97.7	97.3	98.5	88.6	100.9	101.6
35	96.4	95.7	96.2	99.2	98.9	89.4	100.8	101.1
40			92.4	96.8	98.5	88.6	101.1	97.7
45			89.4	97.3	94.4	88.8	99.6	101.2
50			90.3	98.4	95.8	89.1	98.1	96.4
55					92.1	86.4	100.2	96.1
60					93.8	89.4	97.8	93.5
65					92.4	87.6	95.3	98.3
70					93.8	88.6	92.9	91.5
75							96.1	91.4
80							91.9	94.0

Table A3.5.9 Compressor Power Absorbed During Transient Experiments Engine Speed 2600 Rev/Min

TIME (Min)	% STEADY STATE TOTAL USEFUL HEAT OUTPUT (Ambient Enthalpy (kJ/kg dry air)								
	(5)			(10)			(15)		
	Test 1	Test 2	Test 3	Test 1	Test 2	Test 3	Test 1	Test 2	Test 3
0	100	100	100	100	100	100	100	100	100
5	100.3	98.9	99.1	104.2	100	99.4	100	100.6	101.3
10	100	97.9	96.9	101.1	99.8	98.8	100.3	101.3	101.9
15	98.8	97.1	96.2	97.6	97.2	96.7	99.7	100.7	101.8
20	96.9	94.3	93.3	95.9	95.2	93.7	99.5	100.3	100.7
25	93.2	91.8	91.1	95.5	94.5	94.2	99.0	99.6	100
30	89.2	89.0	88.8	99.4	92.2	92.0	98.6	98.9	99.3
35	85.8	85.1	84.5	92.5	91.0	91.0	97.8	98.3	98.9
40	78.1	80.6	80.7	90.8	88.4	88.1	97.4	97.8	98.3
45	77.1	76.9	76.4	88.4	85.5	83.9	96.6	97.1	98.0
50	72.6	71.6	71.3	83.6	82.2	81.6	94.5	95.4	97.1
55	68.3	69.6	66.4	79.3	78.9	76.7	92.8	93.6	94.5
60	63	62	60.6	76.9	75.7	74.6	91.3	92.8	92.5
65				72.2	72.5	70.5	90.0	91.6	90.5
70				68.8	70.1	69.5	88.6	89.6	89.5
75							86.8	87.5	88.4
80							85.1	84.3	85.9
85							82.2	80.6	81.5
90							79.6	77.7	76.0

Table A3.5.10 Calculated Values Of Total Useful Heat Rejected During Transient Experiments Based Upon The Steady State Bowman Heat Recovery Data: Engine Speed 1800 Rev/Min

TIME (Min)	% STEADY STATE TOTAL USEFUL HEAT OUTPUT (Ambient Enthalpy (kJ/kg dry air)							
	(5)		(10)		(15)		(20)	
	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2
0	100	100	100	100	100	100		
5	97.5	99.3	99.7	100	100.1	100.8		
10	96.7	98.0	99.2	99.7	100.1	100.6		
15	94.8	95.4	98.1	98.6	100	100.2		
20	92.4	92.1	97.1	97.6	99.6	99.0		
25	89.4	87.7	95.3	95.5	99.0	97.8		
30	84.2	82.7	91.5	90.9	98.6	97.3	100	100
35	77.2	76.1	87.6	86.9	97.1	96.3	98.7	98.3
40	70.9	69.4	83.4	84.2	95.0	95.3	98.2	96.9
45			78.6	79.4	92.6	93.8	97.2	95.8
50			73.4	74.1	89.3	90.75	96.0	94.3
55			67.3	67.0	86.1	87.3	94.8	92.7
60					83.0	84.5	92.6	91.2
65					79.0	80.4	90.8	89.4
70					74.1	77.3	88.2	87.6
75					67.8	74.2	84.3	84.7
80					61.7	68.6	79.4	80.9
85							73.4	76.4
90							68.4	70.9
95							63.1	63.8

Table A3.5.11 Calculated Values Of Total Useful Heat Rejected During Transient Experiments Based Upon The Steady State Bowman Heat Recovery Data: Engine Speed 2200 Rev/Min

TIME (Min)	% STEADY STATE TOTAL USEFUL HEAT OUTPUT (Ambient Enthalpy (kJ/kg dry air)							
	(5)		(10)		(15)		(20)	
	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2	Test 1	Test 2
0	100	100	100	100	100	100		
5	100.1	98.9	101.4	99.6	99.6	100.7		
10	97.9	96.0	100.2	98.9	99.8	100.4		
15	92.8	92.8	98.9	87.9	99.1	100.2		
20	91.4	89.1	97.4	96.1	98.7	99.4	100	100
25	82.7	82.6	94.9	93.3	96.4	96.9	100.2	99.8
30	76.0	72.8	91.5	89.7	94.4	94.4	99.7	98.9
35	67.0	64.3	86.89	85.2	93.1	92.0	98.8	98.5
40			81.9	79.4	90.1	89.6	96.8	96.1
45			75.4	72.6	85.2	86.6	94.6	93.5
50			65.6	66.5	81.8	83.5	92.0	92.9
55					78.3	81.1	88.9	91.6
60					73.5	76.8	86.8	88.4
65					67.7	69.1	84.7	85.2
70					64.7	63.1	78.8	80.5
75							75.3	75.7
80							61.7	63.6

Table A3.5.12 Calculated Values Of Total Useful Heat Rejected During Transient Experiments Based Upon The Steady State Bowman Heat Recovery Data: Engine Speed 2600 Rev/Min

APPENDIX 3.6

A.G.R. COMPRESSOR PERFORMANCE RESULTS

PARAMETER	TEST 1	TEST 2	TEST 3	TEST 4
Suction Pressure (bar a)	4.3	3.9	4.1	4.0
Suction Temperature (^{o}C)	16.9	17.2	17.2	17.1
Discharge Pressure (bar a)	17.2	16.6	16.4	16.8
Discharge Temperature (^{o}C)	81.1	79.9	79.2	80.7
Compr Power Absorbed (kW)	19.9	19.1	19.2	18.9

Table A3.6.1 A.G.R. Performance With Liquid Injection (2500 Rev/Min)

PARAMETER	TEST 1	TEST 2	TEST 3	TEST 4
Suction Pressure (bar a)	3.8	4.1	4.3	4.0
Suction Temperature (^{o}C)	17.0	16.1	16.6	16.4
Discharge Pressure (bar a)	16.6	16.9	16.7	16.7
Discharge Temperature (^{o}C)	79.1	78.4	79.8	80.2
Compr Power Absorbed (kW)	19.7	19.9	20.3	19.6

Table A3.6.2 A.G.R. Performance With Liquid Injection And Oil Cooling (2500 Rev/Min)

PARAMETER	TEST 1	TEST 2	TEST 3	TEST 4
Suction Pressure (bar a)	4.1	3.8	3.9	4.0
Suction Temperature (^{o}C)	16.2	16.4	15.8	15.5
Discharge Pressure (bar a)	15.5	14.8	15.2	15.2
Discharge Temperature (^{o}C)	97.4	95.0	96.1	98.1
Compr Power Absorbed (kW)	18.1	16.8	17.3	17.8

Table A3.6.3 A.G.R. Performance With Oil Cooling: Liquid Injection Suppressed (2500 Rev/Min)

Suction Conditions

Compressor Pressure	3.16 bar a
Compressor Temperature	2.29 ^{o}C
Specific Enthalpy	255.33 kJ/kg
Density	12.61 kg/m^{3}

Discharge Conditions

Compressor Pressure	18.19 bar a
Compressor Temperature	81.45^{o}C
Specific Enthalpy	294.31 kJ/kg

Interstage Conditions

Compressor Injection Pressure	10.68 bar a
Density Saturated Liquid	1193.81 kg/m^{3}
Specific Enthalpy of Saturated Liquid	75.96 kJ/kg

Evaporating Temperature	-10^{o}C
Condensing Temperature	45^{o}C
Volumetric Flow Rate at Compressor Suction (Based On 80% Volumetric Efficiency [4])	0.0212 m^{3}/s
Measured Power Absorbed	22.8 kW
Heat Rejected to the Condenser Cooling Water	67.9 kW

Table A3.6.4 Typical Set Of Steady State Performance Data: As Used In Calculation Of Liquid Injection Mass Flow Rate

APPENDIX 3.7

STEADY STATE RESULTS:

MODIFIED BOWMAN HEAT RECOVERY EQUIPMENT

Results of the heat recovery experiments are presented graphically, with heat recovery plotted versus engine speed for various fractional engine load factors.

The engine load factors were determined from consideration of the manufacturer's quoted values for full load power.

A mathematical model for the modified Bowman heat recovery equipment based on this data was found to be:

$$\text{Heat Recovery} = (0.0094 + 0.0128E^2)\,N - 9.7 + 18.5E - 15E^2$$

Where:

N = Engine speed (rev/min)

E = Fractional engine load factor

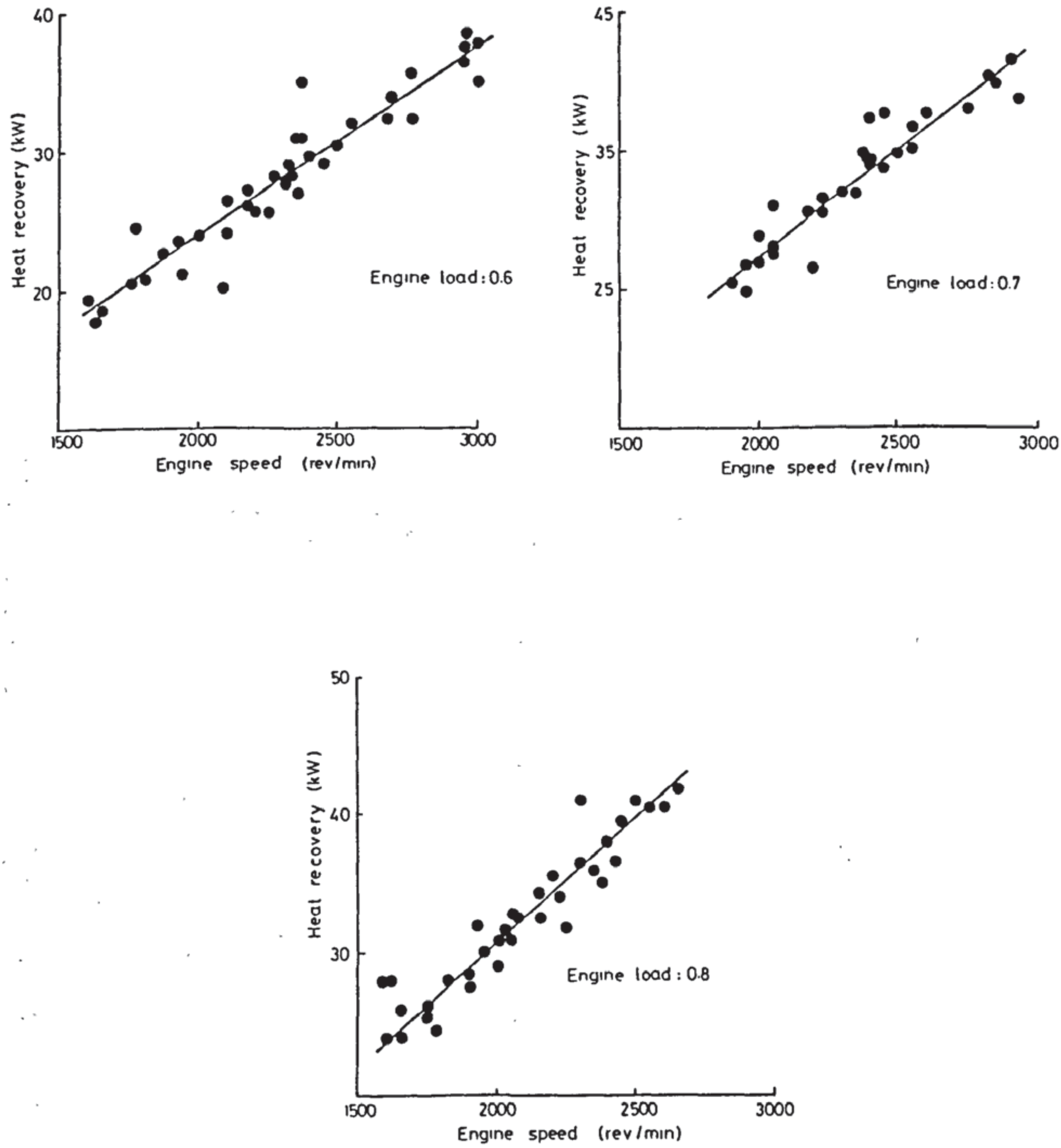

Figure 3.7.1 Steady State Performance Bowman Heat Recovery Equipment

APPENDIX 3.8

FLUIDISED BED TEST RESULTS

Table A3.8.1 Fluidised Bed: Forced Convection Tests

TEST NO. 1

Ambient Temperature	4.5°C
Relative Humidity	92.5 %
Evaporator Air Flow (at commencement of test)	894 cfm
Average Condenser Coolant Temperature	16°C
From Thermodynamic and Transport Properties of Fluids [144]:	
ρ Water	1001 kg/m³
Cp Water	4.186 kJ/kgK

TIME	COMPRESSOR			CONDENSER			AIR TEMP.	
(Min)	Suction Pres. (bar a)	Discharge Pres. (bar a)	Power (kW)	Coolant T (K)	Coolant Flow (m^3/s x10^3)	Heat Rejected (kW)	Onto Evap (°C)	Off Evap (°C)
5	2.31	11.55	1.62	38.3	.0277	4.45	4.4	0.6
10	2.24	10.79	1.62	35.5	.0307	4.57	4.4	0.6
15	2.21	10.59	1.63	35.5	.0309	4.60	4.5	0.4
20	2.24	10.79	1.62	35.5	.0305	4.54	4.5	0.2
25	2.21	11.07	1.62	36.1	.0298	4.51	4.6	0.1
30	2.17	10.59	1.63	35.0	.0308	4.52	4.6	0.0
35	2.14	10.79	1.63	35.0	.03	4.40	4.0	-0.2
40	2.14	11.07	1.65	34.4	.0303	4.37	4.7	-0.3
45	2.11	10.59	1.62	35.6	.0292	4.36	4.7	-0.4
50	2.07	11.07	1.60	34.4	.0296	4.27	4.7	-0.5
55	2.03	11.41	1.59	35.0	.027	3.96	4.6	-0.4
60	2.01	11.41	1.59	34.4	.0271	3.91	4.7	-0.5
65	1.97	11.62	1.58	33.9	.0263	3.74	4.7	-0.6
70	1.83	11.41	1.56	34.6	.0246	3.57	4.7	-0.8

Table A3.8.2 Fluidised Bed: Forced Convection Tests

TEST NO. 2

Ambient Temperature	7.1°C
Relative Humidity	100%
Evaporator Air Flow (at commencement of test)	931 cfm
Average Condenser Coolant Temperature	25°C

From Thermodynamic and Transport Properties of Fluids [144]:

ρ Water	1003 kg/m^3
Cp Water	4.181 kJ/kgK

TIME	COMPRESSOR			CONDENSER			AIR TEMP.	
(Min)	Suction Pres. (bar a)	Discharge Pres. (bar a)	Power (kW)	Coolant T (K)	Coolant Flow (m^3/s x10^3)	Heat Rejected (kW)	Onto Evap (°C)	Off Evap (°C)
5	2.59	12.11		36.5	.0338	5.17	7.0	4.0
10	2.52	11.90		37.8	.0316	5.01	6.8	3.5
15	2.52	11.90		38.6	.0316	5.12	6.8	3.5
20	2.48	11.62		38.3	.0316	5.08	6.7	3.4
25	2.52	11.62		38.9	.0305	4.98	6.6	3.2
30	2.45	10.79		40.7	.0293	5.00	6.9	3.0
35	2.52	11.90		38.8	.0313	5.09	7.0	2.9
40	2.45	11.07		40.8	.0296	5.06	6.9	3.0
45	2.38	10.59		39.7	.03	5.00	7.0	2.9
50	2.38	10.38		39.6	.0298	4.95	7.1	2.8
55	2.31	10.59		39.6	.0291	4.83	7.2	2.7
60	2.24	10.31		41.6	.0274	4.78	7.4	2.2
65	2.21	11.13		36.5	.0292	4.47	7.5	2.7
70	2.14	11.69		36.0	.0279	4.21	7.6	2.6
75	2.13	11.90		36.0	.0276	4.17	7.6	2.4
80	2.04	11.90		35.5	.0263	3.92	7.5	2.3

Table A3.8.3 Fluidised Bed: Fluidised Tests

TEST NO. 3

Ambient Temperature	3.8°C
Relative Humidity	84%
Evaporator Air Flow (at commencement of test)	884 cfm
Average Condenser Coolant Temperature	20°C
From Thermodynamic and Transport Properties of Fluids [144]:	
ρ Water	1001.8 kg/m^3
Cp Water	4.183 kJ/kgK

TIME	COMPRESSOR			CONDENSER			AIR TEMP.	
(Min)	Suction Pres. (bar a)	Discharge Pres. (bar a)	Power (kW)	Coolant T (K)	Coolant Flow (m^3/s x10^3)	Heat Rejected (kW)	Onto Evap (°C)	Off Evap (°C)
0	-	-		-	-	0	3.8	2.8
5	3.14	10.59		12.91	.0602	3.25	3.8	1.6
10	3.09	11.83		37.0	.0299	4.64	3.8	0.9
15	3.04	11.43		35.1	.0352	5.18	3.8	-0.1
20	3.99	13.77		44.1	.0302	5.58	3.8	-0.3
25	3.02	12.03		39.0	.0352	5.75	3.8	-0.4
30	2.98	11.10		35.7	.0415	6.21	3.8	-0.6
40	2.92	11.48		33.6	.0425	5.98	3.8	-0.5
55	2.96	11.10		34.3	.043	6.18	3.8	-0.5
60	2.98	11.48		36.0	.0415	6.26	3.8	-0.4
70	2.91	11.82		38.1	.0386	6.16	3.8	-0.5
80	2.70	12.66		40.3	.0328	5.54	3.8	-0.9
85	2.54	11.43		36.8	.0328	5.06	3.8	-1.5
90	2.37	12.48		35.1	.0298	4.38	3.8	-3.2
95	1.98	11.62		35.3	.025	3.70	3.8	-5.0

APPENDIX 3.9

SPECTRUM ANALYSIS OF SOUND LEVELS EMITTED BY G.E.H.P

INSTALLED FOR A SWIMMING POOL APPLICATION

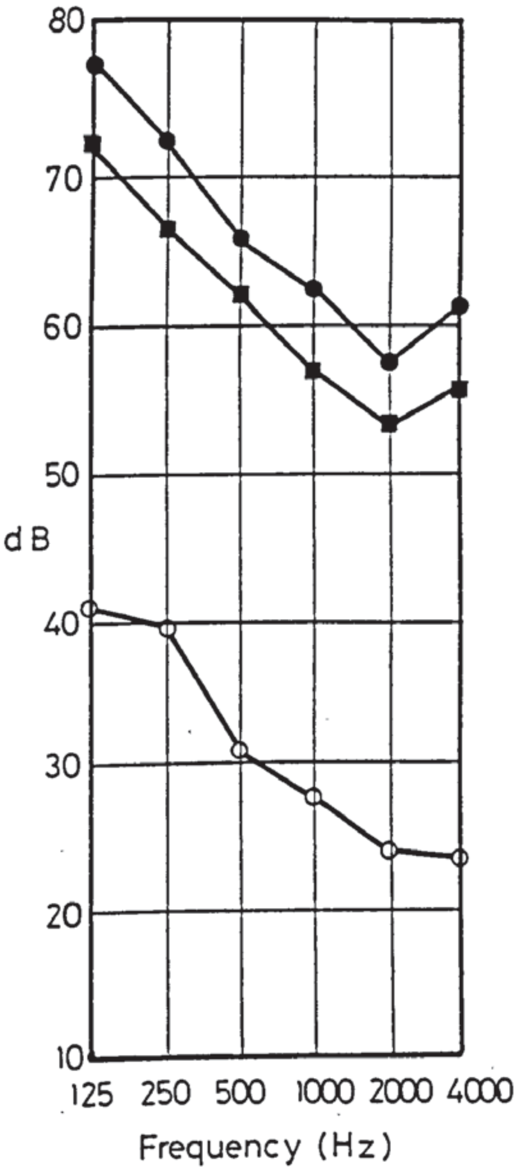

Figure A3.9.1
Noise Spectrum Analysis
Without Attenuators

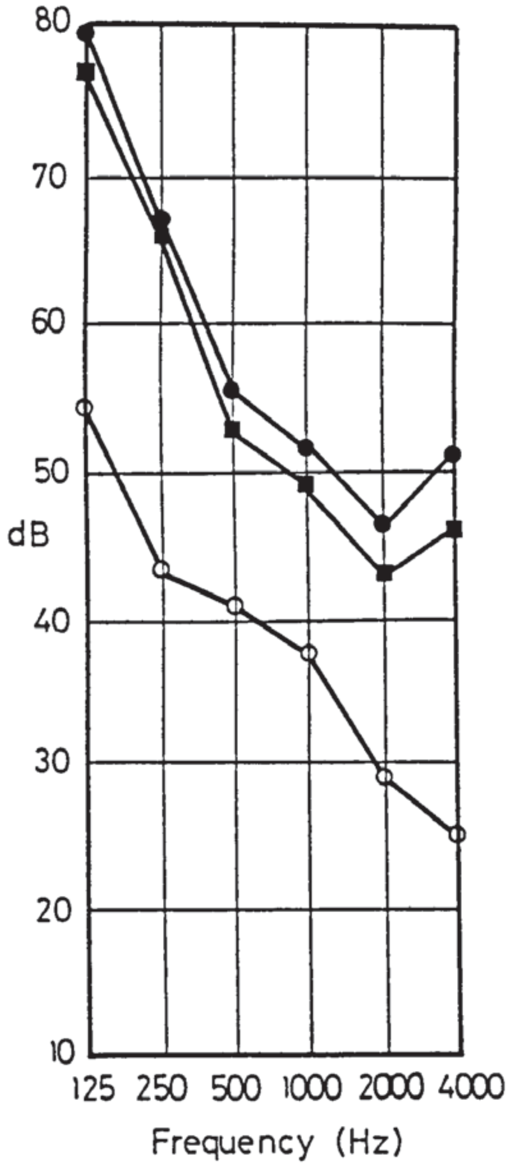

Figure A3.9.2
Noise Spectrum Analysis
With Attenuators

Readings 3m From Plant Room

Complete System Running

System Fans Only

Background Adjacent To Plant Room

APPENDIX 3.10

TABULATED HEAT METERING DATA FOR A SWIMMING POOL APPLICATION

DATE	GAS METER (ft^3)	HEAT METER (MWh)	ENGINE HOURS RUN (h)
1.4.83	540,320	41.4	5631
1.6.83	831,490	142.5	7011.5
1.7.83	918,350	185.3	7695
1.8.83	977,230	205.9	8103
1.9.83	(1)052,950	235.4	8669
1.10.83	(1)178,840	280.5	9189
1.11.83	(1)347,250	334.7	9846
1.12.83	(1)537,280	395.5	10512
1.1.84	(1)740,680	458.4	10998
1.2.84	(1)968,220	532.7	11654
1.3.84	(2)181600	600.5	12193
1.4.84	(3)368740	667.0	12894

Table A3.10.1 Tabulated Heat Metering Data For A Swimming Pool Application

APPENDIX 4

COMPUTER PROGRAMS

APPENDIX 4.1

PROGRAM LISTING FOR G.E.H.P. STEADY STATE TEST ANALYSIS

```
 10 REM   *** CLIVE HICKMAN  -  07-02-1983 ***
 20 REM   *** PROGRAM NAME : HPTESTB ***
 30 REM  *** G.E.H.P. STEADY STATE TEST ANALYSIS ***
 40 REM  *** INPUT DATA AS FOLLOWS ***
 50 REM  ** TEST READINGS IN [mV] **
 60 REM  ** ENGINE SPEED **
 70 REM  ** GAS USED **
 80 REM  ** TEST DURATION [SECS] **
 90 DIM A(6),B(6),C(6),D(6),E(6),F(6),G(6),X(6)
100 OPEN "R",1,"HPTEST.DAT",141
110 FIELD 1,1 AS ZZ$,4 AS A$,4 AS B$,4 AS C$,4 AS D$,4 AS E$,4 AS F$,4 AS G$,4 A
S H$,4 AS J$,4 AS K$,4 AS L$,4 AS M$,4 AS N$,4 AS P$
,4 AS Q$
120 FIELD 1,61 AS DM$,4 AS R$,4 AS S$,4 AS T$,4 AS U$,4 AS V$,4 AS W$,4 AS X$,4
AS Y$,4 AS Z$,4 AS AA$,4 AS AB$,4 AS AC$,4 AS AD$,4
AS AE$,4 AS AF$,4 AS AG$,4 AS AH$,4 AS AJ$,4 AS AK$,4 AS AL$
130 PRINT:PRINT "Program Functions available :":PRINT
140 PRINT "  1 - INITIALISE FILES ."
150 PRINT "  2 - INPUT NEW TEST DATA ."
160 PRINT "  3 - ANALYSE and PRINT TEST RESULTS ."
170 PRINT "  4 - END PROGRAM and RESTORE DATA ."
180 PRINT:INPUT "Which Function do you require ? ",A:IF (A<1)OR(A>4) GOTO 180
190 ON A GOSUB 1910,1980,210,3080
200 GOTO 130
210 REM *** TEST RESULT ANALYSIS ***
220 FOR I=1 TO 6
230 READ A(I)
240 NEXT I
250 FOR I=1 TO 6
260 READ B(I)
270 NEXT I
280 FOR I=1 TO 6
290 READ C(I)
300 NEXT I
310 FOR I=1 TO 6
320 READ D(I)
330 NEXT I
340 FOR I=1 TO 6
350 READ E(I)
360 NEXT I
370 FOR I=1 TO 6
380 READ F(I)
390 NEXT I
400 READ BR
410 READ K1
420 READ D
430 READ HDR
440 READ SDR
450 INPUT "First test number :",M:PRINT
460 INPUT "Second test number:",N:PRINT
470 NUM=1
480 FOR K=M TO N
490 TEST%=K
500 GET 1,TEST%
510 T1=CVS(A$):T2=CVS(B$):T3=CVS(C$):T4=CVS(D$):T5=CVS(E$):T6=CVS(F$)
520 T7=CVS(G$):T8=CVS(H$):T9=CVS(J$):T10=CVS(K$):T11=CVS(L$)
530 T12=CVS(M$):T13=CVS(N$):T14=CVS(P$):T15=CVS(Q$):T16=CVS(R$)
540 T17=CVS(S$):T18=CVS(T$):T19=CVS(U$):T20=CVS(V$):T21=CVS(W$)
550 T22=CVS(X$):T23=CVS(Y$):T24=CVS(Z$):T25=CVS(AA$):T26=CVS(AB$)
```

```
560 T27=CVS(AC$):T28=CVS(AD$):T29=CVS(AE$):T30=CVS(AF$)
570 T31=CVS(AG$):T32=CVS(AH$):T33=CVS(AJ$):T34=CVS(AK$)
580 T35=CVS(AL$)
590 X=24.4583
600 A1=X*((T7-T9-T11)/3)+.25
610 A2=X*((T10-T8-T6)/3)+.25
620 E1=X*T12/3+.25
630 E2=X*(T13)/3+.25
640 SS=X*T17/3+.25
650 AT=X*T14+.25
660 GE=.0296*(10^((7.5*AT)/(237.3+AT)+.78571))
670 EE=GE*T22/2.96
680 IE=(4354*EE)/(1013.25-EE)
690 ATF=AT*1.8+32
700 AE=(ATF*(.241+(.45*IE)/7000)+(1075*IE/7000))*2.326-18.1
710 REM    *** ENTHALPY kJ/kg DRY AIR BASED ON 0 deg.C - 0% R.H. DATUM ***
720 S=X*T5+.25
730 C=X*(-T20)/3+.25
740 C1=X*T21+.25
750 CST=X*T15/3+.25
760 CDT=X*T16/2+.25
770 C2=X*T18/2+.25
780 C3=X*T19/2+.25
790 FC=((T31*10-4)*100/16)/60000!
800 FS=((T32*4.1615-4)*100/16)/60000!
810 P1=T23+.16
820 P2=T24+.05
830 P3=T25+.28
840 P4=T26+.07
850 P5=T27+.03
860 P6=T28-.21
870 CP=6.163*T33*T29/1000
880 IDT=X*T30+.25
890 TR=(E2+273.15)/369.16
900 PR=P5/49.7736
910 GOSUB 2430
920 ET=(TR*369.16)-273.15
930 CPWC=4.187*(.0007214*C1+.8939)
940 CPWS=4.187*(.0007214*SS+.8939)
950 DWC=1.035*(-.0108*(C1)^1.8+1000.47)
960 DWS=1.035*(-.0108*(SS)^1.8+1000.47)
970 CVFUEL=1092
980 QCW=FC*DWC*CPWC*C
990 QSW=FS*DWS*CPWS*SS
1000 QFUEL=T34*CVFUEL/T35
1010 QTW=QCW+QSW
1020 PER=QTW/QFUEL
1030 COPWH=QCW/CP
1040 TR=(C2+273.15)/369.16
1050 PR=P1/49.7736
1060 GOSUB 2430
1070 CT=(TR*369.16)-273.15
1080 TR=(E2+273.15)/369.16
1090 PR=P5/49.7736
1100 GOSUB 2570
1110 H5=HR*9.48491
1120 TR=(CST+273.15)/369.16
1130 PR=P3/49.7736
1140 GOSUB 2570
1150 H3=HR*9.48491
```

```
1160 RDCS=1/(VR*1.90561E-03)
1170  TR=(CDT+273.15)/369.16
1180 PR=P4/49.7736
1190 GOSUB 2570
1200 H4=HR*9.48491
1210 TR=(C2+273.15)/369.16
1220 PR=P1/49.7736
1230 GOSUB 2570
1240 H1=HR*9.48491
1250 TR=(C3+273.15)/369.16
1260 GOSUB 2920
1270 GOSUB 2570
1280 VSV=VR:HSV=HR*9.48491
1290 GOSUB 2970
1300 HEVAP=HE*9.48491
1310 H2=HSV-HEVAP:H6=H2
1320 REM ** REFRIGERANT VOLUMETRIC FLOW IS BASED UPON COMPRESSOR **
1330 REM ** VOLUMETRIC EFFICIENCY OF 85% AS PER EXPERIMENTAL     **
1340 REM ** WORK CARRIED OUT BY K.WOOLAS.THIS IS EQUIVALENT TO   **
1350 REM ** 45cfm AT 3000 rev/min                                **
1360 RMF=(RDCS*45*T33*4.71947E-04)/3000
1370 QCR=RMF*(H1-H2)
1380 QER=RMF*(H5-H6)
1390 QCOMPR=RMF*(H4-H3)
1400 QCDLL=RMF*(H4-H1)
1410 LSC=CT-C3
1420 LPRINT TAB(7) "TEST NUMBER : ";TEST%
1430 LPRINT TAB(7) "~~~~~~~~~~~~~":LPRINT
1440 LPRINT TAB(7) "ENVIRONMENTAL DATA" TAB(46) "PLANT CAPACITIES"
1450 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~" TAB(46) "~~~~~~~~~~~~~~~~"
1460 LPRINT USING "       Engine Speed - rev/min. : ££££.£       ";T33;
1470 LPRINT USING "Condenser Duty - kW.... : £££.££";QCW
1480 LPRINT USING "       Ambient Temp - deg C... : ££££.£       ";AT;
1490 LPRINT USING "Serck Duty ... - kW.... : £££.££";QSW
1500 LPRINT USING "       Ambient R.H. - %....... : ££££.£       ";T22;
1510 LPRINT USING "Total Duty ... - kW.... : £££.££";QTW
1520 LPRINT USING "       Amb. Enthalpy - kJ/kg.. : ££££.£       ";AE;
1530 LPRINT USING "Fuel Consumed  - kW.... : £££.££";QFUEL
1540 LPRINT USING "                                              Power Absorbed -
 kW.... : £££.££";CP :LPRINT
1550 LPRINT TAB(7) "REFRIGERANT CHARACTERISTICS"
1560 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~"
1570 LPRINT USING "       Evap Temp ..... - deg C : £££.££       ";ET;
1580 LPRINT USING "Condensing Temp - deg C : ££.££";CT
1590 LPRINT USING "       Suction Temp .. - deg C : £££.££       ";CST;
1600 LPRINT USING "Condenser Duty  - kW... : ££.££";QCR
1610 LPRINT USING "       Suction Press . - bar.. : £££.££       ";P3;
1620 LPRINT USING "Evaporator Duty - kW... : ££.££";QER
1630 LPRINT USING "       Discharge Temp  - deg C : £££.££       ";CDT;
1640 LPRINT USING "Compressor Duty - kW... : ££.££";QCOMPR
1650 LPRINT USING "       Discharge Press - bar.. : £££.££       ";P4;
1660 LPRINT USING "Heat Lost ..... - kW... : ££.££";QCDLL
1670 LPRINT USING "       Liquid Subcool. - K.... : £££.££       ";LSC:LPRINT:LPRI
NT
1680 LPRINT TAB(7) "PLANT EFFICIENCIES"
1690 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~"
1700 LPRINT USING "       COPh................... : £.££";COPWH
1710 LPRINT USING "       P.E.R.................. : £.££";PER
1720 IF NUM=2 GOTO 1770
1730 LPRINT:LPRINT:LPRINT
```

```
1740 LPRINT "          ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
~~~~~~~~~~~~~~"
1750 LPRINT:LPRINT:
1760 NUM=2:GOTO 1780
1770 LPRINT CHR$(12):NUM=1
1780 NEXT K
1790 DATA 0,-6.401590102,-1.010345093,3.835663598,-1.993351592,1.888547241E+05
1800 DATA 3.742165661,2.378064754,2.468598352,-3.822806065,1.860077264,-1.537861
679E05
1810 DATA 0,-66.42566806,72.25908413,0,-9.643264456,0
1820 DATA 7.030569532,-10.33053763,-7.85605432,3.349627065,0.4458604956,1.032506
622
1830 DATA 9.495332E-02,4.583511,24.42053,-4.6833452,0.0,0.0
1840 DATA 1.6677,1.121739,-.6804608,-.6249227,0.0,0.0
1850 DATA 0.06552
1860 DATA -4.2
1870 DATA 16.733822
1880 DATA 15.303597099
1890 DATA 9.347625337
1900 RETURN
1910 REM  *** INITIALISE FILES ***
1920 PRINT:INPUT "Are you sure ( Y or N ) ? ",AZ$:IF AZ$ <>"Y" THEN RETURN
1930 LSET ZZ$=CHR$(255)
1940 FOR TEST% = 1 TO 1000
1950 PUT 1,TEST%
1960 NEXT TEST%
1970 RETURN
1980 REM  ** INPUT FOR NEW TEST DATA **
1990 INPUT "Test Number  :",TEST% : IF TEST% > 1000 GOTO 2010
2000 GOTO 2020
2010 PRINT "*** TEST NUMBER OUT OF RANGE : NEW FILES REQUIRED ***":PRINT:RETURN
2020 GET 1,TEST%
2030 IF ASC(ZZ$)<>255 THEN INPUT "Is it OK to OVERWRITE this data (Y or N) ?",AZ
$:IF AZ$ <> "Y" THEN RETURN
2040 LSET ZZ$=CHR$(0):PRINT
2050 PRINT "INPUT TEST RESULTS ":PRINT
2060 INPUT "T1 - mV",T1:LSET A$=MKS$(T1)
2070 INPUT "T2 - mV",T2:LSET B$=MKS$(T2)
2080 INPUT "T3 - mV",T3:LSET C$=MKS$(T3)
2090 INPUT "T4 - mV",T4:LSET D$=MKS$(T4)
2100 INPUT "T5 - mV",T5:LSET E$=MKS$(T5)
2110 INPUT "T6 - mV",T6:LSET F$=MKS$(T6)
2120 INPUT "T7 - mV",T7:LSET G$=MKS$(T7)
2130 INPUT "T8 - mV",T8:LSET H$=MKS$(T8)
2140 INPUT "T9 - mV",T9:LSET J$=MKS$(T9)
2150 INPUT "T10 - mV",T10:LSET K$=MKS$(T10)
2160 INPUT "T11 - mV",T11:LSET L$=MKS$(T11)
2170 INPUT "T12 - mV",T12:LSET M$=MKS$(T12)
2180 INPUT "T13 - mV",T13:LSET N$=MKS$(T13)
2190 INPUT "T14 - mV",T14:LSET P$=MKS$(T14)
2200 INPUT "T15 - mV",T15:LSET Q$=MKS$(T15)
2210 INPUT "T16 - mV",T16:LSET R$=MKS$(T16)
2220 INPUT "T17 - mV",T17:LSET S$=MKS$(T17)
2230 INPUT "T18 - mV",T18:LSET T$=MKS$(T18)
2240 INPUT "T19 - mV",T19:LSET U$=MKS$(T19)
2250 INPUT "T20 - mV",T20:LSET V$=MKS$(T20)
2260 INPUT "T21 - mV",T21:LSET W$=MKS$(T21)
2270 INPUT "T22 - mV",T22:LSET X$=MKS$(T22)
2280 INPUT "T23 - mV",T23:LSET Y$=MKS$(T23)
2290 INPUT "T24 - mV",T24:LSET Z$=MKS$(T24)
```

```
2300 INPUT "T25 - mV",T25:LSET AA$=MKS$(T25)
2310 INPUT "T26 - mV",T26:LSET AB$=MKS$(T26)
2320 INPUT "T27 - mV",T27:LSET AC$=MKS$(T27)
2330 INPUT "T28 - mV",T28:LSET AD$=MKS$(T28)
2340 INPUT "T29 - mV",T29:LSET AE$=MKS$(T29)
2350 INPUT "T30 - mV",T30:LSET AF$=MKS$(T30)
2360 INPUT "T31 - mV",T31:LSET AG$=MKS$(T31)
2370 INPUT "T32 - mV",T32:LSET AH$=MKS$(T32)
2380 INPUT "T33 - speed",T33:LSET AJ$=MKS$(T33)
2390 INPUT "T34 - gas used(cu.ft)",T34:LSET AK$=MKS$(T34)
2400 INPUT "T35 - test period(sec)",T35:LSET AL$=MKS$(T35)
2410 PUT 1,TEST%
2420 RETURN
2430 REM ** SUBROUTINE TO EVALUATE SATURATED TEMPERATURE AS A FUNCTION **
2440 REM ** OF PRESSURE FOR REFRIGERANT R22                              **
2450 PRE=D(1)+(D(2)/TR)+(D(3)*LOG(TR))+(D(4)*TR)
2460 IF TR>D(6) GOTO 2480
2470 PRE=PRE+(D(5)*((D(6)/TR)-1)*LOG(D(6)-TR))
2480 PRE=EXP(PRE)
2490 IF ABS(1-PR/PRE)<.000005 GOTO 2560
2500 DPR=((-D(2)/TR+D(3))/TR)+D(4)
2510 IF TR>D(6) GOTO 2530
2520 DPR=DPR-(D(5)*(1+D(6)*LOG(D(6)-TR)/TR)/TR)
2530 DPR=PRE*DPR
2540 TR=TR-((PRE-PR)/DPR)
2550 GOTO 2450
2560 RETURN
2570 REM **SUBROUTINE TO EVALUATE SPECIFIC VOLUME OF VAPOUR FOR R22**
2580 REM              **GIVEN TEMPERATURE AND PRESSURE**
2590 EKT=EXP(K1*TR)
2600 FOR I=1 TO 6
2610 G(I)=A(I)+(B(I)*TR)+(C(I)*EKT)
2620 NEXT I
2630 VRH=.3/1.90561E-03
2640 VRL=.01/1.90561E-03
2650 VR=(VRH+VRL)/2
2660 FOR I=1 TO 6
2670 LET X(I)=(VR-BR)^I
2680 NEXT I
2690 EVDR=EXP(-D*VR)
2700 PRE=G(1)/X(1)+G(2)/X(2)+G(3)/X(3)+G(4)/X(4)+G(5)/X(5)+G(6)*EVDR
2710 IF ABS(1-PRE/PR)<.0000005 GOTO 2790
2720 IF PRE>PR GOTO 2760
2730 VRH=VR
2740 VR=(VR+VRL)/2
2750 GOTO 2660
2760 VRL=VR
2770 VR=(VR+VRH)/2
2780 GOTO 2660
2790 REM ** SUBROUTINE TO EVALUATE SPECIFIC ENTHALPY **
2800 REM ** OF VAPOUR FROM TEMPERATURE AND PRESSURE  **
2810 REM ** AND VOLUME FOR REFRIGERANT R22           **
2820 REM DATA BASE : ENTHALPY OF SATURATED LIQUID AT 223.15K EQUALS 0.0 kJ/kg
2830 EKTR=EXP(K1*TR)
2840 EDVR=EXP(-D*VR)
2850 HRA=HDR-(E(1)/TR)+(E(2)*TR)+(.5*E(3)*TR*TR)+((E(4)*TR^3)/3)
2860 HRB=(A(2)+(C(2)*EKTR*(1-(K1*TR))))/(VR-BR)
2870 HRC=(A(3)+(C(3)*EKTR*(1-(K1*TR))))/(2*(VR-BR)^2)
2880 HRD=(A(5)+(C(5)*EKTR*(1-(K1*TR))))/(4*(VR-BR)^4)
2890 HRE=(A(4)/(3*(VR-BR)^3))+(A(6)*EDVR/D)+(PR*VR)
```

```
2900 HR=HRA+HRB+HRC+HRD+HRE
2910 RETURN
2920 REM *** SUBROUTINE TO EVALUATE SATURATED VAPOUR PRESSURE ***
2930 REM ** FROM SATURATED TEMPERATURE FOR REFRIGERANT R22    **
2940 PR=D(1)+D(2)/TR+D(3)*LOG(TR)+D(4)*TR+(D(5)*((D(6)/TR)-1)*LOG(D(6)-TR))
2950 PR=EXP(PR)
2960 RETURN
2970 REM *** SUBROUTINE TO EVALUATE VOLUME OF SATURATED LIQUID ***
2980 REM *** FROM SATURATED TEMPERATURE FOR REFRIGERANT R22     ***
2990 VSL=1/(1+(F(1)*(1-TR)^.333333)+(F(2)*(1-TR)^.666667)+(F(3)*(1-TR))+(F(4)*(1
-TR)^1.33333))
3000 REM ** SUBROUTINE TO EVALUATE ENTHALPY OF EVAPORATION **
3010 REM ** GIVEN SATURATED TEMPERATURE,SATURATED PRESSURE **
3020 REM ** SPECIFIC VOLUME OF VAPOUR,AND SPECIFIC VOLUME  **
3030 REM ** OF SATURATED LIQUID FOR REFRIGERANT R22        **
3040 REM DATA BASE : ENTHALPY OF SATURATED LIQUID AT 233.15K EQUALS 0.0 kJ/kg
3050 FUNCT=(-D(2)/(TR*TR))+D(3)/TR+D(4)-D(5)/TR-(D(5)*D(6)*LOG(D(6)-TR)/(TR*TR))
3060 HE=PR*TR*(VSV-VSL)*FUNCT
3070 RETURN
3080 CLOSE
3090 PRINT:PRINT " * * * PROGRAM FINISHED * * * "
3100 END
```

Typical Printout From the G.E.H.P. Steady State Test

Analysis Program

TEST NUMBER : 567

ENVIRONMENTAL DATA

Engine Speed - rev/min.	:	2200.0
Ambient Temp - deg C...	:	8.8
Ambient R.H. - %.......	:	59.0
Amb. Enthalpy - kJ/kg..	:	19.3

PLANT CAPACITIES

Condenser Duty - kW....	:	51.74
Serck Duty ... - kW....	:	30.52
Total Duty ... - kW....	:	82.26
Fuel Consumed - kW....	:	61.52
Power Absorbed - kW....	:	17.90

REFRIGERANT CHARACTERISTICS

Evap Temp ..... - deg C	:	-6.23
Suction Temp .. - deg C	:	5.71
Suction Press . - bar..	:	3.69
Discharge Temp - deg C	:	72.28
Discharge Press - bar..	:	17.15
Liquid Subcool. - K....	:	4.50

Condensing Temp - deg C	:	43.51
Condenser Duty - kW...	:	39.92
Evaporator Duty - kW...	:	35.98
Compressor Duty - kW...	:	7.15
Heat Lost ..... - kW...	:	4.95

PLANT EFFICIENCIES

COPh....................	:	2.89
P.E.R...................	:	1.34

TEST NUMBER : 568

ENVIRONMENTAL DATA

Engine Speed - rev/min.	:	2200.0
Ambient Temp - deg C...	:	9.1
Ambient R.H. - %.......	:	59.4
Amb. Enthalpy - kJ/kg..	:	19.8

PLANT CAPACITIES

Condenser Duty - kW....	:	52.05
Serck Duty ... - kW....	:	31.13
Total Duty ... - kW....	:	83.18
Fuel Consumed - kW....	:	61.52
Power Absorbed - kW....	:	18.03

REFRIGERANT CHARACTERISTICS

Evap Temp ..... - deg C	:	-5.94
Suction Temp .. - deg C	:	6.20
Suction Press . - bar..	:	3.75
Discharge Temp - deg C	:	73.50
Discharge Press - bar..	:	17.28
Liquid Subcool. - K....	:	4.63

Condensing Temp - deg C	:	43.89
Condenser Duty - kW...	:	40.50
Evaporator Duty - kW...	:	36.24
Compressor Duty - kW...	:	7.42
Heat Lost ..... - kW...	:	5.20

PLANT EFFICIENCIES

COPh....................	:	2.89
P.E.R...................	:	1.35

Data From Test 568 Is Used In Figure 4.12, Page 77

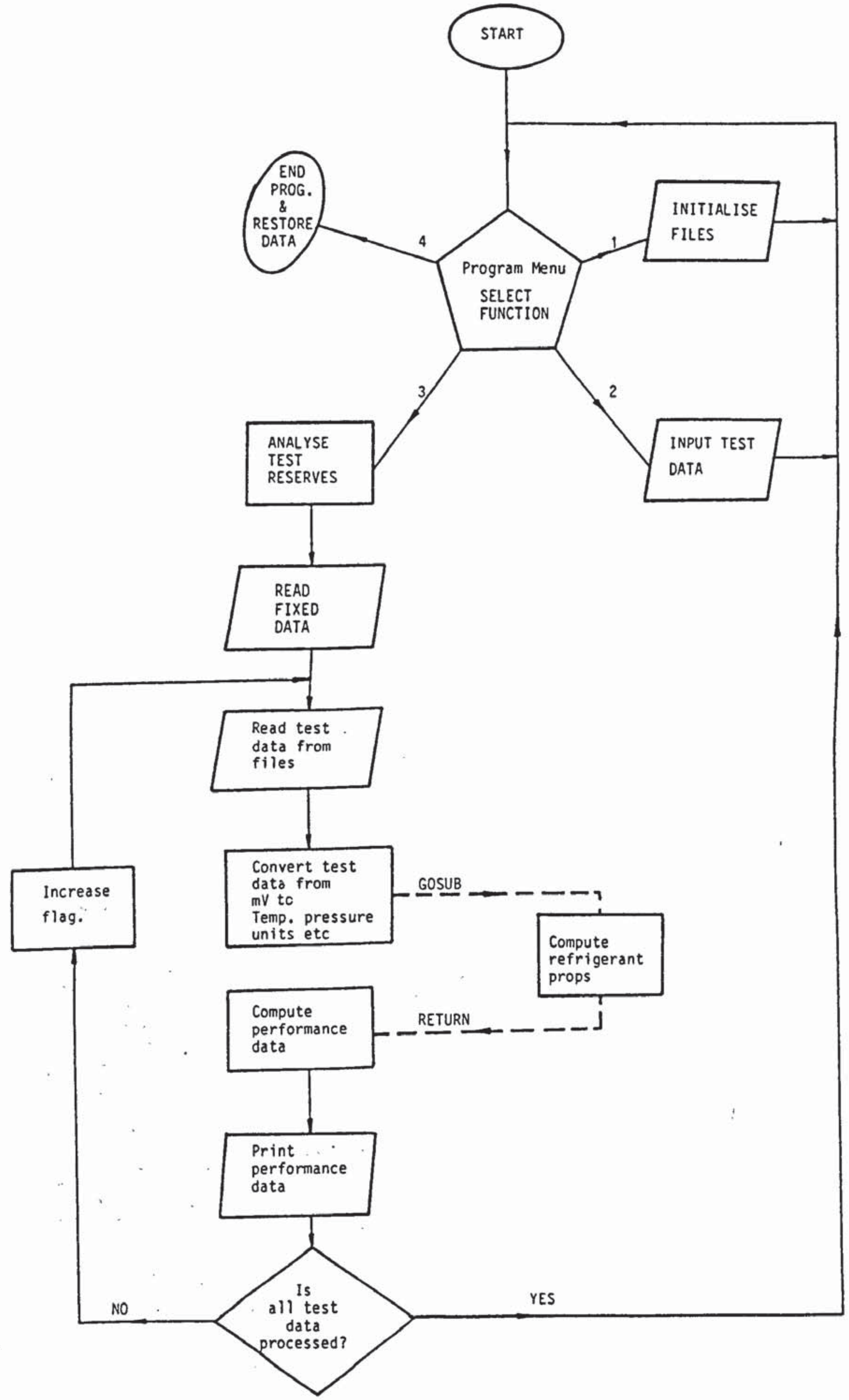

Figure A4.1.1 Flow Chart For G.E.H.P. Steady State Result Analysis

APPENDIX 4.2

PROGRAM LISTING FOR G.E.H.P. TRANSIENT TEST ANALYSIS

```
10 REM   *** CLIVE HICKMAN  -  25-10-1983 ***
20 REM   *** PROGRAM NAME : HPTRAN ***
30 REM  *** G.E.H.P. TRANSIENT TEST ANALYSIS ***
40 REM  *** INPUT DATA AS FOLLOWS ***
50 REM  ** TEST READINGS IN [mV] **
60 REM  ** ENGINE SPEED **
70 DIM A(6),B(6),C(6),D(6),E(6),F(6),G(6),X(6)
80 OPEN "R",1,"HPTRAN.DAT",61
90 FIELD 1,1 AS ZZ$,4 AS A$,4 AS B$,4 AS C$,4 AS D$,4 AS E$,4 AS F$,4 AS G$,4 AS
 H$,4 AS J$,4 AS K$,4 AS L$,4 AS M$,4 AS N$,4 AS P$,
4 AS Q$
100 PRINT:PRINT "Program Functions available :":PRINT
110 PRINT "  1 - INITIALISE FILES ."
120 PRINT "  2 - INPUT NEW TEST DATA ."
130 PRINT "  3 - ANALYSE and PRINT TEST RESULTS ."
140 PRINT "  4 - END PROGRAM and RESTORE DATA ."
150 PRINT:INPUT "Which Function do you require ? ",A:IF (A<1)OR(A>4) GOTO 150
160 ON A GOSUB 1220,1290,180,1680
170 GOTO 100
180 REM *** TEST RESULT ANALYSIS ***
190 FOR I=1 TO 6
200 READ A(I)
210 NEXT I
220 FOR I=1 TO 6
230 READ B(I)
240 NEXT I
250 FOR I=1 TO 6
260 READ C(I)
270 NEXT I
280 FOR I=1 TO 6
290 READ D(I)
300 NEXT I
310 FOR I=1 TO 6
320 READ E(I)
330 NEXT I
340 FOR I=1 TO 6
350 READ F(I)
360 NEXT I
370 READ BR
380 READ K1
390 READ D
400 READ HDR
410 READ SDR
420 INPUT "First test number :",M:PRINT
430 INPUT "Second test number:",N:PRINT
440 LPRINT "GEHP TRANSIENT PERFORMANCE":LPRINT:LPRINT:LPRINT
450 LPRINT"TIME          COND.DUTY     TOTAL DUTY     EVAP TEMP     COND TEMP
 COMP POWER"
460 LPRINT"mins             %              %              %              %
    %     ":LPRINT
470 TIME=0
480 FOR K=M TO N
490 TEST%=K
500 GET 1,TEST%
510 T1=CVS(A$):T2=CVS(B$):T3=CVS(C$):T4=CVS(D$):T5=CVS(E$):T6=CVS(F$)
520 T7=CVS(G$):T8=CVS(H$):T9=CVS(J$):T10=CVS(K$):T11=CVS(L$)
530 T12=CVS(M$):T13=CVS(N$):T14=CVS(P$):T15=CVS(Q$)
540 X=24.4583
550 E2=X*(T2)/3+.25
560 SS=X*T4/3+.25
```

```
570 AT=X*T3+.25
580 GE=.0296*(10^((7.5*AT)/(237.3+AT)+.78571))
590 EE=GE*T8/2.96
600 IE=(4354*EE)/(1013.25-EE)
610 ATF=AT*1.8+32
620 AE=(ATF*(.241+(.45*IE)/7000)+(1075*IE/7000))*2.326-18.1
630 REM   *** ENTHALPY kJ/kg DRY AIR BASED ON 0 deg.C - 0% R.H. DATUM ***
640 S=X*T1+.25
650 C=X*T6/3+.25
660 C1=X*T7+.25
670 C2=X*T5/2+.25
680 FC=((T13*10-4)*100/16)/60000!
690 FS=((T14*4.1615-4)*100/16)/60000!
700 P1=T9+.16
710 P5=T10+.03
720 CP=6.163*T15*T11/1000
730 IDT=X*T12+.25
740 TR=(E2+273.15)/369.16
750 PR=P5/49.7736
760 GOSUB 1540
770 ET=(TR*369.16)-273.15
780 CPWC=4.187*(.0007214*C1+.8939)
790 CPWS=4.187*(.0007214*SS+.8939)
800 DWC=1.035*(-.0108*(C1)^1.8+1000.47)
810 DWS=1.035*(-.0108*(SS)^1.8+1000.47)
820 QCW=FC*DWC*CPWC*C
830 QSW=FS*DWS*CPWS*SS
840 QTW=QCW+QSW
850 TR=(C2+273.15)/369.16
860 PR=P1/49.7736
870 GOSUB 1540
880 CT=(TR*369.16)-273.15
890 IF K=M GOTO 910
900 GOTO 960
910 QCWK=QCW
920 QTWK=QTW
930 ETK=AT-ET
940 CTK=CT
950 CPK=CP
960 QCW=(QCW/QCWK)*100
970 QTW=(QTW/QTWK)*100
980 ET=(EKT/(AT-ET))*100
990 CT=(CT/CTK)*100
1000 CP=(CP/CPK)*100
1010 LPRINT TIME,     QCW,     QTW,     ET,     CT,     CP
1020 ATS=ATS+AT:AES=AES+AE:ST8=ST8+T8
1030 TIME=TIME+5
1040 NEXT K
1050 ATA=ATS/(N-M+1):AEA=AES/(N-M+1):AT8=ST8/(N-M+1)
1060 LPRINT:LPRINT:LPRINT "ENGINE SPEED FOR TEST (rpm) ............. : ";T15
1070 LPRINT:LPRINT "AVERAGE AMBIENT TEMP FOR TEST (deg C).... : ";ATA
1080 LPRINT:LPRINT "AVERAGE AMBIENT HUMIDITY FOR TEST (R.H) . : ";AT8
1090 LPRINT:LPRINT "AVERAGE AMBIENT ENTHALPY FOR TEST (kJ/kg) : ";AEA
1100 DATA 0,-6.401590102,-1.010345093,3.835663598,-1.993351592,1.888547241E+05
1110 DATA 3.742165661,2.378064754,2.468598352,-3.822806065,1.860077264,-1.537861
679E05
1120 DATA 0,-66.42566806,72.25908413,0,-9.643264456,0
1130 DATA 7.030569532,-10.33053763,-7.85605432,3.349627065,0.4458604956,1.032506
622
1140 DATA 9.495332E-02,4.583511,24.42053,-4.6833452,0.0,0.0
```

```
1150 DATA 1.6677,1.121739,-.6804608,-.6249227,0.0,0.0
1160 DATA 0.06552
1170 DATA -4.2
1180 DATA 16.733822
1190 DATA 15.303597099
1200 DATA 9.347625337
1210 RETURN
1220 REM  *** INITIALISE FILES ***
1230 PRINT:INPUT "Are you sure ( Y or N ) ? ",AZ$:IF AZ$ <>"Y" THEN RETURN
1240 LSET ZZ$=CHR$(255)
1250 FOR TEST% = 1 TO 1000
1260 PUT 1,TEST%
1270 NEXT TEST%
1280 RETURN
1290 REM  ** INPUT FOR NEW TEST DATA **
1300 INPUT "Test Number  :",TEST% : IF TEST% > 1000 GOTO 1320
1310 GOTO 1330
1320 PRINT "*** TEST NUMBER OUT OF RANGE : NEW FILES REQUIRED ***":PRINT:RETURN
1330 GET 1,TEST%
1340 IF ASC(ZZ$)<>255 THEN INPUT "Is it OK to OVERWRITE this data (Y or N) ?",AZ
$:IF AZ$ <> "Y" THEN RETURN
1350 LSET ZZ$=CHR$(0):PRINT
1360 PRINT "INPUT TEST RESULTS ":PRINT
1370 INPUT "T1 - mV",T1:LSET A$=MKS$(T1)
1380 INPUT "T2 - mV",T2:LSET B$=MKS$(T2)
1390 INPUT "T3 - mV",T3:LSET C$=MKS$(T3)
1400 INPUT "T4 - mV",T4:LSET D$=MKS$(T4)
1410 INPUT "T5 - mV",T5:LSET E$=MKS$(T5)
1420 INPUT "T6 - mV",T6:LSET F$=MKS$(T6)
1430 INPUT "T7 - mV",T7:LSET G$=MKS$(T7)
1440 INPUT "T8 - mV",T8:LSET H$=MKS$(T8)
1450 INPUT "T9 - mV",T9:LSET J$=MKS$(T9)
1460 INPUT "T10 - mV",T10:LSET K$=MKS$(T10)
1470 INPUT "T11 - mV",T11:LSET L$=MKS$(T11)
1480 INPUT "T12 - mV",T12:LSET M$=MKS$(T12)
1490 INPUT "T13 - mV",T13:LSET N$=MKS$(T13)
1500 INPUT "T14 - mV",T14:LSET P$=MKS$(T14)
1510 INPUT "T15 - speed",T15:LSET Q$=MKS$(T15)
1520 PUT 1,TEST%
1530 RETURN
1540 REM ** SUBROUTINE TO EVALUATE SATURATED TEMPERATURE AS A FUNCTION **
1550 REM ** OF PRESSURE FOR REFRIGERANT R22                            **
1560 PRE=D(1)+(D(2)/TR)+(D(3)*LOG(TR))+(D(4)*TR)
1570 IF TR>D(6) GOTO 1590
1580 PRE=PRE+(D(5)*((D(6)/TR)-1)*LOG(D(6)-TR))
1590 PRE=EXP(PRE)
1600 IF ABS(1-PR/PRE)<.000005 GOTO 1670
1610 DPR=((-D(2)/TR+D(3))/TR)+D(4)
1620 IF TR>D(6) GOTO 1640
1630 DPR=DPR-(D(5)*(1+D(6)*LOG(D(6)-TR)/TR)/TR)
1640 DPR=PRE*DPR
1650 TR=TR-((PRE-PR)/DPR)
1660 GOTO 1560
1670 RETURN
1680 CLOSE
1690 PRINT:PRINT " * * * PROGRAM FINISHED * * * "
1700 END
```

APPENDIX 4.3

PROGRAM LISTING FOR THE GAS ENGINE HEAT RECOVERY EQUIPMENT STEADY STATE ANALYSIS

```
10 REM    *** CLIVE HICKMAN : 5:11:83 ***
20 REM    *** PROGRAM NAME   : BO-MAN  ***
30 REM    *** ANALYSIS OF BOWMAN HEAT RECOVERY EQUIPMENT ***
40 DIM F(6),A(6),B(6),C(6),E(2),X(6)
50 FOR I=1 TO 6
60 READ A(I)
70 NEXT I
80 FOR I=1 TO 6
90 READ B(I)
100 NEXT I
110 FOR I=1 TO 6
120 READ C(1)
130 NEXT I
140 READ BR
150 READ K1
160 READ D
170 LPRINT "BOWMAN PERFORMANCE "
180 LPRINT "~~~~~~~~~~~~~~~~~~ ":LPRINT
190 LPRINT " SPEED              POWER          HEAT REC.        FUEL"
200 LPRINT "  rpm          `       kW              kW              kW "
210 LPRINT
220 INPUT "No.Sets Test Results : ";N:PRINT
230 FOR J=1 TO N
240 INPUT " Gas Flow (kW)              :   ";GF
250 INPUT " Engine Speed (rpm)         :   ";SPEED
260 INPUT " Water On Temp (Deg.C)      :   ";TON
270 INPUT " Water Off Temp (Deg.C)     :   ";TOFF
280 INPUT " Water Flow Rate (m3/s)     :   ";FLOW
290 INPUT " Compressor Power (mV)      : ";CP
300 CPWATER =4.187*((.0007214*TON)+.8939)
310 DENWATER =1.035*(-.0108*(TON)^1.8+1000.47)
320 HTREC=FLOW*DENWATER*CPWATER*(TOFF-TON)
330 QCOMP=6.163*CP*SPEED/1000
340 LPRINT SPEED,QCOMP,HTREC,GF
350 NEXT J
360 END
```

APPENDIX 4.4

A) PROGRAM LISTING FOR THE G.E.H.P. COMPUTER DESIGN AID: REFRIGERANT R22

```
10 REM *** CLIVE HICKMAN : 16.6.83 ***
20 REM *** PROGRAM NAME : GEHP-22 ***
30 DIM EVAP(10),QE(10),POWER(10),QC(10),LOA(10),FUEL(10),EXHAUST(10),JACKET(10),
HEATREC(10),QT(10),PER(10),COP(10),ETF(10)
40 PRINT:PRINT:PRINT
50 PRINT "GEHP SELECTION PROGRAM "
60 LPRINT "                              GEHP SELECTION PROGRAM"
70 PRINT "~~~~~~~~~~~~~~~~~~~~~~"
80 LPRINT "                              ~~~~~~~~~~~~~~~~~~~~~~"
90 PRINT "ALL DATA WILL BE OUTPUT TO PRINTER":PRINT
100 COUNTER=1
110 PRINT "Design Conditions :":PRINT
120 LPRINT "        Design Conditions :"
130 LPRINT "        ~~~~~~~~~~~~~~~~~~"
140 INPUT "Evaporating Temperature ...Deg C :",ET
150 LPRINT "        Evaporating Temperature ... Deg C : ";ET
160 INPUT "Condensing Temperature ....Deg C :",CT
170 LPRINT "        Condensing Temperature .... Deg C : ";CT
180 INPUT "Heat Load Requirement  ....kW    :",Q
190 LPRINT "        Heat Load Requirement ..... kW    : ";Q
200 INPUT "Operating Speed ...........rpm (0 if unknown)   :",N:PRINT
210 IF N=0 GOTO 230
220 GOTO 240
230 N=2600
240 LPRINT "        Operating Speed ........... rpm   : ";N:LPRINT
250 PRINT "Input Anticipated Range Of Evaporating Temperature"
260 PRINT "Input design condition if unknown"
270 LPRINT TAB(7) "Anticipated range of evaporating temperature "
280 INPUT "Maximum Evap. Temp. .......Deg C :",ET1
290 LPRINT TAB(7) "Maximum evaporating temperature ....deg C : ";ET1
300 INPUT "Minimum Evap. Temp. .......Deg C :",ET2:PRINT
310 LPRINT TAB(7) "Minimum evaporating temperature ....deg C : ";ET2:LPRINT
320 IF CT<=55 THEN PRINT "Refrigerant R22 Recommended ":PRINT
330 IF CT>80 THEN PRINT "Maximum Condensing Temperature Exceeded ":PRINT:GOTO 35
30
340 IF CT >55 THEN PRINT "Refrigerant R12 Recommended" : PRINT "PROGRAM FOR R12
DATA BEING LOADED AND RUN":PRINT:GOTO 360
350 GOTO 370
360 CHAIN"GEHP-12",10,ALL
370 CTF=CT*1.8+32:ETF=ET*1.8+32:Z=0
380 REM *** R22 SYSTEM SELECTION ***
390 REM *** FOR R12 SELECTION DATA SEE PROGRAM GEHP-12 ***
400 REM *** CALCULATION OF DESIGN DUTY FOR FORD/AGR(R22) UNIT ***
410 QE=(95.951-6.651*SQR(CTF)+(ETF*(.8854-4.47E-06*CTF*CTF))+((ETF*ETF)*(.00701-
1.53E-07*CTF*CTF)))/.95
420 POWER=((3.95+.1672*CTF)+(ETF*(3.02*CTF^(-.877)))+((ETF*ETF)*(.0041-.0266/SQR
(CTF))))/1.0526
430 QC=(QE+POWER)*N/3000
440 POWER=POWER*N/3000
450 QE=QE*N/3000
460 LOA=(-.0006823+108.804/N)*POWER^(1/(.9872+120350!/(N*N)))
470 IF LOA>1! THEN 550
480 IF LOA <.4 GOTO 550
490 FUEL=(33.8461*LOA)-16.645+(N*(.012885+(.01538*LOA)))
500 JACKET=24.295-(3.4E+07/(N*N))-10*(1-LOA)
510 EXHAUST=((9.4614*LOA)-10.445+N*(.004845+(.01007*LOA)))*.8
520 QT1=QC+JACKET+EXHAUST:QT=QT1
530 IF QT1>=Q THEN Z=1
540 IF QT1>=Q GOTO 1960
550 REM *** DETERMINATION OF SCREW COMPRESSOR SERIES (R22) ***
```

```
560 IF ET2<-40 THEN LPRINT "Minimum Evaporating Temperature Too Low For Screw Co
mpressors":GOTO 3530
570 IF ET1>12 THEN LPRINT "Maximum Evaporating Temperature To High For Screw Com
pressors":GOTO 3530
580 ETMAX=-39.3485+.988395*CT-4.79843E-03*CT*CT
590 ETMIN=-57.0482+.0110248*CT*CT
600 IF ET2<ETMIN THEN LPRINT "Minimum Evaporating Temperature Too Low For Screw
Compressors":GOTO 3530
610 IF ET1<ETMAX THEN 1540
620 ETMAX=-31.4211+1.02597*CT-5.24572E-03*CT*CT
630 ETMIN=-55.2331+.842385*CT-3.93157E-03*CT*CT
640 IF ET2<ETMIN THEN LPRINT "Evaporating Temperature Range Too Wide For Screw C
ompressors":GOTO 3530
650 IF ET1<ETMAX THEN 1130
660 ETMAX=-36.8863+8.48768*SQR(CT)
670 ETMIN=-56.2836+6.73102*SQR(CT)
680 IF ET2<ETMIN THEN LPRINT "Evaporating Temperature Range Too Wide For Screw C
ompressors":GOTO 3530
690 IF ET1>ETMAX THEN LPRINT "Maximum Evaporating Temperature Too High For Screw
 Compressors":GOTO 3530
700 REM *** CALCULATIONS OF DESIGN DUTY FOR OSK(R22) COMPRESSORS ***
710 REM *** CALCULATIONS OF DESIGN DUTY FOR LEYLAND-BITZER(OSK-R22) UNIT  ***
720 QE=(117.654-.0131136*CT*CT+ET*(4.36907-.0233833*CT)+ET*ET*(.0416931-9.50507E
-07*CT*CT))*N/2900
730 POWER=(14.3256*EXP(.0158865*CT)+ET*(-.470141+.0403936*CT-5.80193E-04*CT*CT)+
ET*ET*(.0221077-1.09345E-03*CT+1.42378E-05*CT*CT))*N
/2900
740 QC=QE+POWER
750 LOA =POWER/(1/(.0215537+92694.6/(N*N)))
760 IF LOA > 1! GOTO 830
770 IF LOA < .4 GOTO 830
780 FUEL=(-17.5+.0475*N)*LOA
790 JACKET=(-12.5+.01875*N)*LOA
800 EXHAUST=(-10+.02*N)*LOA*.8
810 QT2=QC+JACKET+EXHAUST:QT=QT2
820 IF QT2>=Q GOTO 2160
830 REM *** CALCULATION OF DESIGN DUTY FOR SI4-BITZER(OSK-R22) UNIT ***
840 QE=(201.669-.0224523*CT*CT+ET*(7.759*EXP(-6.93097E-03*CT))+ET*ET*(.0529152+9
.40454E-04*CT-1.3079E-05*CT*CT))*N/2900
850 POWER=(23.3789*EXP(.0159058*CT)+ET*(-.674009+.0614168*CT-8.92191E-04*CT*CT)+
ET*ET*(.0296441-.0014712*CT+1.94758E-05*CT*CT))*N/29
00
860 QC=QE+POWER
870 LOA=POWER/((-327.771+52.1602* LOG(N))*.67)
880 IF LOA >1! THEN 950
890 IF LOA < .4 THEN 950
900 FUEL=.67*(34.2451+143.121*LOA-190.19*LOA*LOA-(2.30198E-03-.139836*LOA*LOA)*N
)
910 JACKET=.67*(29.4375-17.1687*LOA-(7.63154E-03-.0279542*LOA)*N)
920 EXHAUST=.67*(27.0529-54.0081*LOA*LOA-(4.00125E-03-.0695442*LOA*LOA)*N)*.8
930 QT3=QC+EXHAUST+JACKET:QT=QT3
940 IF QT3>=Q THEN 2200
950 REM *** CALCULATION OF DESIGN DUTY FOR SI6-BITZER(OSK-R22) UNIT ***
960 QE=(230.512-.025659*CT*CT+ET*(8.86503*EXP(-.0069223*CT))+ET*ET*(.0598509+1.1
0951E-03*CT-1.54127E-05*CT*CT))*N/2900
970 POWER=(28.0708+.291597*CT+6.51261E-03*CT*CT+ET*(-.777698+.0705718*CT-1.02457
E-03*CT*CT)+ET*ET*(.0341873-1.69842E-03*CT+2.24907E-
05*CT*CT))*N/2900
980 QC=QE+POWER
990 LOA=POWER/(-327.771+52.1602*LOG(N))
```

```
1000 IF LOA > 1! THEN PRINT "Engine Load Factor Too High":GOTO 3530
1010 IF LOA < .4 THEN PRINT "Engine Load Factor Too Low":GOTO 3530
1020 FUEL=34.2451+143.121*LOA-190.19*LOA*LOA-(2.30198E-03-.139836*LOA*LOA)*N
1030 JACKET=29.4375-17.1687*LOA-(7.63154E-03-.0279542*LOA)*N
1040 EXHAUST=27.0529-54.0081*LOA*LOA-(4.00125E-03-.0695442*LOA*LOA)*N*.8
1050 QT4=QC+EXHAUST+JACKET:QT=QT4
1060 IF QT>=Q THEN 2240
1070 GOSUB 1090
1080 GOTO 1120
1090 IF COUNTER=1 GOTO 5100
1100 LPRINT "The required performance is outside the range of this program"
1110 RETURN
1120 GOTO 3530
1130 REM   *** CALCULATION OF DESIGN DUTY FOR OSN(R22) SERIES COMPRESSORS ***
1140 REM   *** CALCULATION OF DESIGN DUTY FOR LEYLAND-BITZER(DSN-R22) MODULE ***
1150 QE=(117.167*EXP(-1.32284E-04*CT*CT)+ET*(3.45777-2.28065E-04*CT*CT)+ET*ET*(.
0187586+5.22748E-04*CT-6.93698E-06*CT*CT))*N/2900
1160 POWER=(21.012+4.33817E-03*CT*CT+ET*(-.0940287+.0246783*CT-3.31949E-04*CT*CT
)+ET*ET*(.0157948-8.36951E-04*CT+1.20307E-05*CT*CT))
*N/2900
1170 QC=QE+POWER
1180 LOA=POWER/(1/(.0215537+92694.6/(N*N)))
1190 IF LOA>1 THEN 1260
1200 IF LOA<.4 THEN 1260
1210 FUEL=LOA*(-17.5+.0475*N)
1220 JACKET=LOA*(.01875*N-12.5)
1230 EXHAUST=LOA*(.02*N-10)*.8
1240 QT2=QC+JACKET+EXHAUST:QT=QT2
1250 IF QT>=Q THEN 3560
1260 REM   *** CALCULATION OF DESIGN DUTY FOR SI4-BITZER(DSN-R22) MODULE ***
1270 QE=(194.703-.019586*CT*CT+ET*(5.93071-3.89824E-04*CT*CT)+ET*ET*(.0303979+9.
76563E-04*CT-1.27456E-05*CT*CT))*N/2900
1280 POWER=(33.6625+.0073151*CT*CT+ET*(-.771639+.0657038*CT-7.90893E-04*CT*CT)+E
T*ET*(.0127269-8.39413E-04*CT+1.45952E-05*CT*CT))*N/
2900
1290 QC=QE+POWER
1300 LOA=POWER/(.67*(-327.771+52.1602*LOG(N)))
1310 IF LOA>1 THEN 1380
1320 IF LOA<.4 THEN 1380
1330 FUEL=.67*(34.2451+143.121*LOA-190.19*LOA*LOA-N*(2.30198E-03-.139836*LOA*LOA
))
1340 JACKET=.67*(29.4375-17.1687*LOA-N*(7.63154E-03-.0279542*LOA))
1350 EXHAUST=.67*(27.0529-54.0081*LOA*LOA-N*(4.00125E-03-.0695442*LOA*LOA))*.8
1360 QT3=QC+EXHAUST+JACKET:QT=QT3
1370 IF QT3>Q THEN 3600
1380 REM   *** CALCULATION OF DESIGN DUTY FOR SI6-BITZER(DSN-R22) MODULE ***
1390 QE=N/2900*(222.501-.0223779*CT*CT+ET*(6.77677-4.45117E-04*CT*CT)+ET*ET*(.03
448+1.12838E-03*CT-1.47074E-05*CT*CT))
1400 POWER=N/2900*(39.2332+.0081036*CT*CT+ET*(-.186714+.0464483*CT-6.22866E-04
T*CT)+ET*ET*(.0305562-1.61902E-03*CT+2.31478E-05*CT*
CT))
1410 QC=QE+POWER
1420 LOA=POWER/(-327.771+52.1602*LOG(N))
1430 IF LOA>1 THEN PRINT"Engine Load Factor Too High":GOTO 3530
1440 IF LOA<.4 THEN PRINT"Engine Load Factor Too Low":GOTO 3530
1450 FUEL=34.2451+143.121*LOA-190.19*LOA*LOA-N*(2.30198E-03-.139836*LOA*LOA)
1460 JACKET=29.4375-17.1687*LOA-N*(7.63154E-03-.0279542*LOA)
1470 EXHAUST=27.0529-54.0081*LOA*LOA-N*(4.00125E-03-.0695442*LOA*LOA)*.8
1480 QT4=QC+JACKET+EXHAUST:QT=QT4
1490 IF QT>=Q THEN 3640
```

```
1500 GOSUB 1090
1510 GOTO 3530
1520 GOSUB 1090
1530 GOTO 3530
1540 REM   *** CALCULATION OF DESIGN DUTY FOR OST(R22) SERIES COMPRESSORS ***
1550 REM   *** CALCULATION OF DESIGN DUTY FOR LEYLAND-BITZER(OST-R22) MODULE ***
1560 QE=N/2900*(106.675-.010601*CT*CT+ET*(2.99372-2.3473E-04*CT*CT)+ET*ET*(7.779
56E-03+.0772004/SQR(CT)))
1570 POWER=N/2900*(24.7105*EXP(1.38362E-04*CT*CT)+ET*(.123714+.0161436*CT-1.9356
9E-04*CT*CT)+ET*ET*(.0100931-4.22015E-04*CT+5.41751E
-06*CT*CT))
1580 QC=QE+POWER
1590 LOA=POWER/(1/(.0215537+92694.6 /(N*N)))
1600 IF LOA>1 THEN 1670
1610 IF LOA<.4 THEN 1670
1620 FUEL=LOA*(.0475*N-17.5)
1630 JACKET=LOA*(.01875*N-12.5)
1640 EXHAUST=LOA*(.02*N-10)*.8
1650 QT2=QC+EXHAUST+JACKET:QT=QT2
1660 IF QT>=Q THEN 3680
1670 REM   *** CALCULATION OF DESIGN DUTY FOR SI4-BITZER(OST-R22) MODULE ***
1680 QE=N/2900*(183.098-.0182107*CT*CT+(5.13522-3.99106E-04*CT*CT)*ET+(.0246087+
.388347/CT)*ET*ET)
1690 POWER=N/2900*(1/(.0288694-2.28865E-04*CT)+ET*(.21764+.0255252*CT-3.02201E-0
4*CT*CT)+ET*ET*(.0181172-7.74941E-04*CT+1.00144E-05*
CT*CT))
1700 QC=QE+POWER
1710 LOA=POWER/(.67*(-327.771+52.1602*LOG(N)))
1720 IF LOA>1 THEN 1790
1730 IF LOA<.4 THEN 1790
1740 FUEL=.67*(34.2451+143.121*LOA-190.19*LOA*LOA-N*(2.30198E-03-.139836*LOA*LOA
))
1750 JACKET=.67*(29.4375-17.1687*LOA-N*(7.63154E-03-.0279542*LOA))
1760 EXHAUST=.67*(27.0529-54.0081*LOA*LOA-N*(4.00125E-03-.0695442*LOA*LOA))*.8
1770 QT3=QC+JACKET+EXHAUST:QT=QT3
1780 IF QT3>=Q THEN 3720
1790 REM   *** CALCULATION OF DESIGN DUTY FOR SI6-BITZER(OST-R22) MODULE ***
1800 QE=N/2900*(211.598-.0214001*CT*CT+ET*(4.57826+.0803571*CT-1.42627E-03*CT*CT
)+ET*ET*(.0375+7.5581E-04*CT-1.21686E-05*CT*CT))
1810 POWER=N/2900*(1/(.0252415-1.99918E-04*CT)+ET*(.25905+.0288734*CT-3.43643E-0
4*CT*CT)+ET*ET*(.0207755-8.85471E-04*CT+1.14072E-05*
CT*CT))
1820 QC=QE+POWER
1830 LOA=POWER/(-327.771+52.1602*LOG(N))
1840 IF LOA>1 THEN PRINT"Engine Load Factor Too High":GOTO 3530
1850 IF LOA<.4 THEN PRINT"Engine Load Factor Too Low":GOTO 3530
1860 FUEL=34.2451+143.121*LOA-190.19*LOA*LOA-N*(2.30198E-03-.139836*LOA*LOA)
1870 JACKET=29.4375-17.1687*LOA-N*(7.63154E-03-.0279542*LOA)
1880 EXHAUST=27.0529-54.0081*LOA*LOA-N*(4.00125E-03-.0695442*LOA*LOA)*.8
1890 QT4=QC+JACKET+EXHAUST:QT=QT4
1900 IF QT>=Q THEN 3760
1910 GOSUB 1090
1920 GOTO 3530
1930 GOSUB 1090
1940 GOTO 3530
1950 REM   *** PERFORMANCE DATA BASED ON DESIGN CONDITIONS (R22) ***
1960 IF COUNTER=2 THEN GOSUB 5130
1970 LPRINT "        Optimum Unit : FORD KENT-AGR450 R22 Module"
1980 LPRINT "        ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
1990 LPRINT USING  "        TOTAL SYSTEM DUTY (kW) : £££.££";QT:LPRINT:LPRINT
```

```
2000 LPRINT "        Performance Characteristics"
2010 LPRINT "        ~~~~~~~~~~~~~~~~~~~~~~~~~~~"
2020 LPRINT "        Refrigerant Type .................... :  R22"
2030 LPRINT USING"           Evaporator Duty (kW) ................ : £££.££";QE
2040 LPRINT USING "          Power Absorbed (kW) ................. : £££.££";POWER
2050 LPRINT USING "          Condenser Duty (kW) ................. : £££.££";QC:LOA
=LOA*100
2060 LPRINT USING "          Engine Loading (%) .................. : £££.££";LOA
2070 LPRINT USING "          Fuel Usage (kW) ..................... : £££.££";FUEL
2080 LPRINT USING "          Exhaust Heat Recovery (kW) .......... : £££.££";EXHAUS
T
2090 LPRINT USING "          Jacket Water Heat Recovery (kW) ..... : £££.££";JACKET
2100 LPRINT USING "          Total System Duty (kW) .............. : £££.££";QT :PE
R=QT/FUEL
2110 LPRINT USING "          Primary Energy Ratio (P.E.R.) ....... : ££.££";PER :CO
P=QC/POWER
2120 LPRINT USING "          Coefficient Of Performance (C.O.P.) . : ££.££";COP :LP
RINT
2130 INPUT "Is Performance Over The Full Range Of Evaporating Temperature Requir
ed (YorN)?",A$
2140 IF A$="Y" THEN 2280
2150 IF A$<>"Y" THEN 3530
2160 IF COUNTER=2 THEN GOSUB 5130
2170 LPRINT TAB(7) "Optimum Unit : LEYLAND-BITZER 6061 OSK-R22 Module"
2180 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
2190 GOTO 1990
2200 IF COUNTER=2 THEN GOSUB 5130
2210 LPRINT TAB(7) "Optimum Unit : SI4-BITZER 7051 OSK-R22 Module"
2220 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
2230 GOTO 1990
2240 IF COUNTER=2 THEN GOSUB 5130
2250 LPRINT TAB(7) "Optimum Unit : SI6-BITZER 7061 OSK-R22 Module"
2260 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
2270 GOTO 1990
2280 REM *** CALCULATION OF UNIT DUTIES OVER A RANGE OF EVAP. TEMPS. (R22) ***
2290 REM *** FOR FORD-AGR AND BITZER OSK SERIES UNITS ***
2300 J=((ET1-ET2)\5)+1
2310 EVAP(1)=ET2
2320 IF Q<=QT1 THEN 2940
2330 IF Q<=QT2 THEN 2740
2340 IF Q<=QT3 THEN 2540
2350 REM   *** SI6-BITZER(OSK) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R22) ***
2360 FOR I=1 TO J
2370 QE(I)=(230.512-.025659*CT*CT+EVAP(I)*(8.86506*EXP(-.0069223*CT))+EVAP(I)*EV
AP(I)*(.0598509+1.10951E-03*CT-1.54127E-05*CT*CT))*N
/2900
2380 POWER(I)=(28.0708+.291597*CT+6.51261E-03*CT*CT+EVAP(I)*(-.777698+.0705718*C
T-1.02457E-03*CT*CT)+EVAP(I)*EVAP(I)*(.0341873-1.698
42E-03*CT+2.24907E-05*CT*CT))*N/2900
2390 QC(I)=QE(I)+POWER(I):LOA(I)=POWER(I)/(-327.771+52.1602*LOG(N))
2400 IF LOA(I)<.4 THEN 2500
2410 IF LOA(I)>1! THEN 2500
2420 FUEL(I)=34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)+N*(-2.30198E-03+.139836
*LOA(I)*LOA(I))
2430 JACKET(I)=29.4375-17.1687*LOA(I)+N*(-7.63154E-03+.0279542*LOA(I))
2440 EXHAUST(I)=27.0529-54.0081*LOA(I)*LOA(I)+N*(-4.00125E-03+.0695442*LOA(I)*LO
A(I))*.8
2450 HEATREC(I)=JACKET(I)+EXHAUST(I)
2460 QT(I)=QC(I)+HEATREC(I)
2470 PER(I)=QT(I)/FUEL(I)
```

```
2480 COP(I)=QC(I)/POWER(I)
2490 GOTO 2510
2500 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
2510 EVAP(I+1)=EVAP(I)+5
2520 NEXT I
2530 GOTO 3160
2540 REM *** S14-BITZER(DSK) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R22) ***
2550 FOR I=1 TO J
2560 QE(I)=(201.669-.0224523*CT*CT+EVAP(I)*(7.759*EXP(-6.93097E-03*CT))+EVAP(I)*
EVAP(I)*(.0529152+9.40454E-04*CT-1.3079E-05*CT*CT))*
N/2900
2570 POWER(I)=(23.3789*EXP(.0159058*CT)+EVAP(I)*(-.674009+.0614168*CT-8.92191E-0
4*CT*CT)+EVAP(I)*EVAP(I)*(.0296441-.0014714*CT+1.947
58E-05*CT*CT))*N/2900
2580 QC(I)=QE(I)+POWER(I)
2590 LOA(I)=POWER(I)/(.67*(-327.771+52.1602*LOG(N)))
2600 IF LOA(I)<.4 THEN 2700
2610 IF LOA(I)>1 THEN 2700
2620 FUEL(I)=.67*(34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)-N*(2.30198E-03-.13
9836*LOA(I)*LOA(I)))
2630 JACKET(I)=.67*(29.4375-17.1687*LOA(I)-N*(7.63154E-03-.0279542*LOA(I)))
2640 EXHAUST(I)=.67*(27.0529-54.0081*LOA(I)*LOA(I)-N*(4.00125E-03-.0695442*LOA(I
)*LOA(I)))*.8
2650 QT(I)=QC(I)+JACKET(I)+EXHAUST(I)
2660 HEATREC(I)=EXHAUST(I)+JACKET(I)
2670 PER(I)=QT(I)/FUEL(I)
2680 COP(I)=QC(I)/POWER(I)
2690 GOTO 2710
2700 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
2710 EVAP(I+1)=EVAP(I)+5
2720 NEXT I
2730 GOTO 3190
2740 REM *** LEYLAND-BITZER(DSK) PERFORMANCE OVER RANGE OF EVAP. TEMP.(R22) ***
2750 FOR I=1 TO J
2760 QE(I)=(117.654-.0131136*CT*CT+EVAP(I)*(4.36907-.0233833*CT)+EVAP(I)*EVAP(I)
*(.0416931-9.50507E-07*CT*CT))*N/2900
2770 POWER(I)=(14.3256*EXP(.0158865*CT)+EVAP(I)*(-.470141+.0403936*CT-5.80193E-0
4*CT*CT)+EVAP(I)*EVAP(I)*(.0221077-1.09345E-03*CT+1.
42378E-05*CT*CT))*N/2900
2780 QC(I)=QE(I)+POWER(I)
2790 LOA(I)=POWER(I)/(1/(.0215537+92694.6/(N*N)))
2800 IF LOA(I)<.4 THEN 2900
2810 IF LOA(I)>1 THEN 2900
2820 FUEL(I)=LOA(I)*(-17.5+.0475*N)
2830 JACKET(I)=LOA(I)*(-12.5+.01875*N)
2840 EXHAUST(I)=LOA(I)*(.02*N-10)*.8
2850 HEATREC(I)=EXHAUST(I)+JACKET(I)
2860 QT(I)=QC(I)+HEATREC(I)
2870 PER(I)=QT(I)/FUEL(I)
2880 COP(I)=QC(I)/POWER(I)
2890 GOTO 2910
2900 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
2910 EVAP(I+1)=EVAP(I)+5
2920 NEXT I
2930 GOTO 3220
2940 REM *** FORD-AGR PERFORMANCE OVER RANGE OF EVAP.TEMP.(R22) ***
2950 FOR I=1 TO J
```

```
2960 ETF(I)=EVAP(I)*1.8+32
2970 QE(I)=(N/3000)*(95.951-6.651*SQR(CTF)+(ETF(I)*(.8854-4.47E-06*CTF*CTF))+((E
TF(I)*ETF(I))*(.00701-1.53E-07*CTF*CTF)))/.95
2980 POWER(I)=(N/3000)*((3.95+.1672*CTF)+(ETF(I)*(3.02*CTF^(-.877)))+((ETF(I)*ET
F(I))*(.0041-.0266/(SQR(CTF)))))/1.0526
2990 QC(I)=QE(I)+POWER(I)
3000 LOA(I)=(-.0006823+108.804/N)*POWER(I)^(1/(.9872+120350!/(N*N)))
3010 IF LOA(I)<.4 THEN 3110
3020 IF LOA(I)>1 THEN 3110
3030 FUEL(I)=(33.8461*LOA(I))-16.654+(N*(.012885+(.01538*LOA(I))))
3040 JACKET(I)=24.295-(3.4E+07/(N*N))-10*(1-LOA(I))
3050 EXHAUST(I)=((9.4614*LOA(I))-10.445+N*(.004845+(.01007*LOA(I))))*.8
3060 HEATREC(I)=JACKET(I)+EXHAUST(I)
3070 QT(I)=QC(I)+HEATREC(I)
3080 PER(I)=QT(I)/FUEL(I)
3090 COP(I)=QC(I)/POWER(I)
3100 GOTO 3120
3110 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
3120 EVAP(I+1)=EVAP(I)+5
3130 NEXT I
3140 GOTO 3250
3150 REM    *** TABULATED PERFORMANCE DATA ***
3160 LPRINT TAB(7) "Tabulated Data For SI6-BitzerOSK Module :"
3170 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~";LPRINT
3180 GOTO 3390
3190 LPRINT TAB(7) "Tabulated Data For SI4-BitzerOSK Module :"
3200 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~";LPRINT
3210 GOTO 3390
3220 LPRINT TAB(7)"Tabulated Data For Leyland-BitzerOSK Module :"
3230 LPRINT TAB(7)"~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~";LPRINT
3240 GOTO 3390
3250 LPRINT TAB(7)"Tabulated Data For Ford-OGK Module :"
3260 LPRINT TAB(7)"~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~";LPRINT:GOTO 3390
3270 LPRINT TAB (7) "TABULATED DATA FOR SI6-BITZER(OSN) MODULE "
3280 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~ ":LPRINT:GOTO 3390
3290 LPRINT TAB(7) "TABULATED DATA FOR SI4-BITZER(OSN) MODULE "
3300 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~ ":LPRINT:GOTO 3390
3310 LPRINT TAB(7) "TABULATED DATA FOR LEYLAND-BITZER(OSN) MODULE "
3320 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~ ":LPRINT:GOTO
3390
3330 LPRINT TAB(7) "TABULATED DATA FOR SI6-BITZER(OST) MODULE "
3340 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~ ":LPRINT:GOTO 3390
3350 LPRINT TAB(7) "TABULATED DATA FOR SI4-BITZER(OST) MODULE "
3360 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~ ":LPRINT:GOTO 3390
3370 LPRINT TAB(7) "TABULATED DATA FOR LEYLAND-BITZER(OST) MODULE "
3380 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~ ":LPRINT:GOTO
3390
3390 LPRINT "        Refrigerant Type ..................... :  R22 "
3400 LPRINT"        Condensing Temperature (Deg C) ...... : ";CT:LPRINT
3410 LPRINT"          Evap   Power    Eng     Condr   Heat    Total    Fuel     CO
P    PER"
3420 LPRINT "         Temp     Abs     Load    Duty    Rec     Duty     Usage"
3430 LPRINT "          (C)     (kW)            (kW)    (kW)    (kW)     (kW)"
3440 LPRINT
3450 FOR I=1 TO J
3460 LPRINT USING "          £££.£   ££.£     £.££   £££.£   £££.£   £££.£   £££.
£    £.££    £.££";EVAP(I);POWER(I);LOA(I);QC(I);HEAT
REC(I);QT(I);FUEL(I);COP(I);PER(I)
3470 NEXT I
```

```
3480 IF Z=1 THEN 3530
3490 LPRINT:LPRINT:LPRINT
3500 LPRINT TAB(7) "Limits of operation ":LPRINT
3510 LPRINT USING "        Max. evaporating temperature deg C : £££.££";ETMAX
3520 LPRINT USING "        Min. evaporating temperature deg C : £££.££";ETMIN
3530 INPUT "Are Any Other Checks Required (YorN)? :",A$
3540 IF A$="Y" THEN PRINT"DF":GOTO 50
3550 IF A$<>"Y" THEN 5150
3560 IF COUNTER=2 THEN GOSUB 5130
3570 LPRINT TAB(7) "Optimum Unit : LEYLAND-BITZER 6061 OSN-R22 Module"
3580 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3590 GOTO 1990
3600 IF COUNTER=2 THEN GOSUB 5130
3610 LPRINT TAB(7) "Optimum Unit : SI4-BITZER 7051 OSN-R22 Module"
3620 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3630 GOTO 1990
3640 IF COUNTER=2 THEN GOSUB 5130
3650 LPRINT TAB(7) "Optimum Unit SI6-BITZER 7061OSN-R22 Module"
3660 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3670 GOTO 1990
3680 IF COUNTER=2 THEN GOSUB 5130
3690 LPRINT TAB(7) "Optimum Unit :LEYLAND-BITZER 6061 OST-R22 Module"
3700 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3710 GOTO 1990
3720 IF COUNTER=2 THEN GOSUB 5130
3730 LPRINT TAB(7) "Optimum Unit : SI4-BITZER 7051 OST-R22 Module"
3740 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3750 GOTO 1990
3760 IF COUNTER=2 THEN GOSUB 5130
3770 LPRINT TAB(7) "Optimum Unit : SI6-BITZER OST-R22 Module"
3780 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3790 GOTO 1990
3800 REM *** CALCULATION OF UNIT DUTIES OVER A RANGE OF EVAP. TEMPS. (R22) ***
3810 REM *** FOR BITZER OSN SERIES UNITS ***
3820 J=((ET1-ET2)\5)+1
3830 EVAP(1)=ET2
3840 IF Q<=QT2 THEN 4250
3850 IF Q<=QT3 THEN 4050
3860 REM   *** SI6-BITZER(OSN) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R22) ***
3870 FOR I= 1 TO J
3880 QE(I)=(222.501-.0223779*CT*CT+EVAP(I)*(6.77677-4.45117E-04*CT*CT)+EVAP(I)*E
VAP(I)*(.03448+1.12838E-03*CT-1.47074E-05*CT*CT))*N/
2900
3890 POWER(I)=(39.2332+.0081036*CT*CT+EVAP(I)*(-.186714+.0464483*CT-6.22866E-04*
CT*CT)+EVAP(I)*EVAP(I)*(.0305562-1.61902E-03*CT+2.31
478E-05*CT*CT))*N/2900
3900 QC(I)=QE(I)+POWER(I):LOA(I)=POWER(I)/(-327.771+52.1602*LOG(N))
3910 IF LOA(I)<.4 THEN 4010
3920 IF LOA(I)>1! THEN 4010
3930 FUEL(I)=34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)+N*(-2.30198E-03+.139836
*LOA(I)*LOA(I))
3940 JACKET(I)=29.4375-17.1687*LOA(I)+N*(-7.63154E-03+.0279542*LOA(I))
3950 EXHAUST(I)=27.0529-54.0081*LOA(I)*LOA(I)+N*(-4.00125E-03+.0695442*LOA(I)*LO
A(I))*.8
3960 HEATREC(I)=JACKET(I)+EXHAUST(I)
3970 QT(I)=QC(I)+HEATREC(I)
3980 PER(I)=QT(I)/FUEL(I)
3990 COP(I)=QC(I)/POWER(I)
4000 GOTO 4020
4010 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
```

```
C(I)=0
4020 EVAP(I+1)=EVAP(I)+5
4030 NEXT I
4040 GOTO 3270
4050 REM *** SI4-BITZER(DSN) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R22) ***
4060 FOR I=1 TO J
4070 QE(I)=(194.703-.019586*CT*CT+EVAP(I)*(5.93071-3.89824E-04*CT*CT)+EVAP(I)*EV
AP(I)*(.0303979+9.76563E-04*CT-1.27456E-05*CT*CT))*N
/2900*.8
4080 POWER(I)=(33.6625+.0073151*CT*CT+EVAP(I)*(-.771639+.0657038*CT-7.90893E-04*
CT*CT)+EVAP(I)*EVAP(I)*(.127269-8.39413E-04*CT+1.459
52E-05*CT*CT))*N/2900
4090 QC(I)=QE(I)+POWER(I)
4100 LOA(I)=POWER(I)/(.67*(-327.771+52.1602*LOG(N)))
4110 IF LOA(I)<.4 THEN 4210
4120 IF LOA(I)>1 THEN 4210
4130 FUEL(I)=.67*(34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)-N*(2.30198E-03-.13
9836*LOA(I)*LOA(I)))
4140 JACKET(I)=.67*(29.4375-17.1687*LOA(I)-N*(7.63154E-03-.0279542*LOA(I)))
4150 EXHAUST(I)=.67*(27.0529-54.0081*LOA(I)*LOA(I)-N*(4.00125E-03-.0695442*LOA(I
)*LOA(I)))*.8
4160 QT(I)=QC(I)+JACKET(I)+EXHAUST(I)
4170 HEATREC(I)=JACKET(I)+EXHAUST(I)
4180 PER(I)=QT(I)/FUEL(I)
4190 COP(I)=QC(I)/POWER(I)
4200 GOTO 4220
4210 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
4220 EVAP(I+1)=EVAP(I)+5
4230 NEXT I
4240 GOTO 3290
4250 REM *** LEYLAND-BITZER(DSN) PERFORMANCE OVER RANGE OF EVAP. TEMP.(R22) ***
4260 FOR I=1 TO J
4270 QE(I)=(117.167*EXP(-1.32284E-04*CT*CT)+EVAP(I)*(3.45777-2.28065E-04*CT*CT)+
EVAP(I)*EVAP(I)*(.0187586+5.22748E-04*CT-6.93698E-06
*CT*CT))*N/2900
4280 POWER(I)=(21.012+4.33817E-03*CT*CT+EVAP(I)*(-.0940287+.0246783*CT-3.31949E-
04*CT*CT)+EVAP(I)*EVAP(I)*(.0157948-8.36951E-04*CT+1
.20307E-05*CT*CT))*N/2900
4290 QC(I)=QE(I)+POWER(I)
4300 LOA(I)=POWER(I)/(1/(.0215537+92694.6/(N*N)))
4310 IF LOA(I)<.4 THEN 4410
4320 IF LOA(I)>1 THEN 4410
4330 FUEL(I)=LOA(I)*(-17.5+.0475*N)
4340 JACKET(I)=LOA(I)*(-12.5+.01875*N)
4350 EXHAUST(I)=LOA(I)*(.02*N-10)*.8
4360 HEATREC(I)=EXHAUST(I)+JACKET(I)
4370 QT(I)=QC(I)+HEATREC(I)
4380 PER(I)=QT(I)/FUEL(I)
4390 COP(I)=QC(I)/POWER(I)
4400 GOTO 4420
4410 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
4420 EVAP(I+1)=EVAP(I)+5
4430 NEXT I
4440 GOTO 3310
4450 REM *** CALCULATION OF UNIT DUTIES OVER A RANGE OF EVAP. TEMPS. (R22) ***
4460 REM *** FOR BITZER OST SERIES UNITS ***
4470 J=((ET1-ET2)\5)+1
4480 EVAP(1)=ET2
```

```
4490 IF Q<=QT2 THEN 4900
4500 IF Q<=QT3 THEN 4700
4510 REM   *** SI6-BITZER(OST) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R22) ***
4520 FOR I=1 TO J
4530 QE(I)=(211.598-.0214001*CT*CT+EVAP(I)*(4.57826+.0803571*CT-1.42627E-03*CT*C
T)+EVAP(I)*EVAP(I)*(.0375+7.5581E-04*CT-1.21686E-05*
CT*CT))*N/2900
4540 POWER(I)=(1/(.0252415-1.99918E-04*CT)+EVAP(I)*(.25905+.0288734*CT-3.43643E-
04*CT*CT)+EVAP(I)*EVAP(I)*(.0207755-8.85471E-04*CT+1
.14072E-05*CT*CT))*N/2900
4550 QC(I)=QE(I)+POWER(I):LOA(I)=POWER(I)/(-327.771+52.1602*LOG(N))
4560 IF LOA(I)<.4 THEN 4660
4570 IF LOA(I)>1! THEN 4660
4580 FUEL(I)=34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)+N*(-2.30198E-03+.139836
*LOA(I)*LOA(I))
4590 JACKET(I)=29.4375-17.1687*LOA(I)+N*(-7.63154E-03+.0279542*LOA(I))
4600 EXHAUST(I)=27.0529-54.0081*LOA(I)*LOA(I)+N*(-4.00125E-03+.0695442*LOA(I)*LO
A(I))*.8
4610 HEATREC(I)=JACKET(I)+EXHAUST(I)
4620 QT(I)=QC(I)+HEATREC(I)
4630 PER(I)=QT(I)/FUEL(I)
4640 COP(I)=QC(I)/POWER(I)
4650 GOTO 4670
4660 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
4670 EVAP(I+1)=EVAP(I)+5
4680 NEXT I
4690 GOTO 3330
4700 REM *** SI4-BITZER(OST) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R22) ***
4710 FOR I=1 TO J
4720 QE(I)=(183.098-.0182107*CT*CT+EVAP(I)*(5.13522-3.99106E-04*CT*CT)+EVAP(I)*E
VAP(I)*(.0246087+.388347/CT))*N/2900
4730 POWER(I)=(1/(.0288694-2.28865E-04*CT)+EVAP(I)*(.21764+.0255252*CT-3.02201E-
04*CT*CT)+EVAP(I)*EVAP(I)*(.0181172-7.74941E-04*CT+1
.00144E-05*CT*CT))*N/2900
4740 QC(I)=QE(I)+POWER(I)
4750 LOA(I)=POWER(I)/(.67*(-327.771+52.1602*LOG(N)))
4760 IF LOA(I)<.4 THEN 4860
4770 IF LOA(I)>1 THEN 4860
4780 FUEL(I)=.67*(34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)-N*(2.30198E-03-.13
9836*LOA(I)*LOA(I)))
4790 JACKET(I)=.67*(29.4375-17.1687*LOA(I)-N*(7.63154E-03-.0279542*LOA(I)))
4800 EXHAUST(I)=.67*(27.0529-54.0081*LOA(I)*LOA(I)-N*(4.00125E-03-.0695442*LOA(I
)*LOA(I)))*.8
4810 HEATREC(I)=JACKET(I)+EXHAUST(I)
4820 QT(I)=QC(I)+JACKET(I)+EXHAUST(I)
4830 PER(I)=QT(I)/FUEL(I)
4840 COP(I)=QC(I)/POWER(I)
4850 GOTO 4870
4860 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
4870 EVAP(I+1)=EVAP(I)+5
4880 NEXT I
4890 GOTO 3350
4900 REM *** LEYLAND-BITZER(OST) PERFORMANCE OVER RANGE OF EVAP. TEMP.(R22) ***
4910 FOR I=1 TO J
4920 QE(I)=(106.675-.010601*CT*CT+EVAP(I)*(2.99372-2.3473E-04*CT*CT)+EVAP(I)*EVA
P(I)*(7.77956E-03+.0772004/SQR(CT)))*N/2900
4930 POWER(I)=(24.7105*EXP(1.38362E-04*CT*CT)+EVAP(I)*(.123714+.0161436*CT-1.935
69E-04*CT*CT)+EVAP(I)*EVAP(I)*(.0100913-4.22015E-04*
```

```
CT+5.41751E-06*CT*CT))*N/2900
4940 QC(I)=QE(I)+POWER(I)
4950 LOA(I)=POWER(I)/(1/(.0215537+92694.6/(N*N)))
4960 IF LOA(I)<.4 THEN 5060
4970 IF LOA(I)>1 THEN 5060
4980 FUEL(I)=LOA(I)*(-17.5+.0475*N)
4990 JACKET(I)=LOA(I)*(-12.5+.01875*N)
5000 EXHAUST(I)=LOA(I)*(.02*N-10)*.8
5010 HEATREC(I)=EXHAUST(I)+JACKET(I)
5020 QT(I)=QC(I)+HEATREC(I)
5030 PER(I)=QT(I)/FUEL(I)
5040 COP(I)=QC(I)/POWER(I)
5050 GOTO 5070
5060 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
5070 EVAP(I+1)=EVAP(I)+5
5080 NEXT I
5090 GOTO 3370
5100 Q=Q/2
5110 COUNTER=2
5120 GOTO 320
5130 LPRINT TAB(7)  "TWO MODULES REQUIRED : PERFORMANCE OF SINGLE MODULE FOLLOWS
 :":LPRINT
5140 RETURN
5150 END
```

B) PROGRAM LISTING FOR THE G.E.H.P. COMPUTER DESIGN AID

REFRIGERANT R12

```
10 REM *** CLIVE HICKMAN : 16.6.83 ***
20 REM *** PROGRAM NAME : GEHP-12 ***
30 CTF=CT*1.8+32:ETF=ET*1.8+32:Z=0
40 REM *** R12 SYSTEM SELECTION ***
50 REM *** FOR R22 SELECTION DATA SEE PROGRAM GEHP-22 ***
60 REM *** CALCULATION OF DESIGN DUTY FOR FORD/AGR(R12) UNIT ***
70 QE=37.4243+(.96648*ETF)+(.002298*ETF*ETF)+((-.17383+(.003237*ETF))*CTF)
80 POWER=10.025*EXP((.002206+.0000256*ETF)*CTF)/1.0526
90 QC=(QE+POWER)*N/3000
100 POWER=POWER*N/3000
110 QE=QE*N/3000
120 LOA=(-.0006823+108.804/N)*POWER^(1/(.9872+120350!/(N*N)))
130 IF LOA>1! THEN 210
140 IF LOA <.4 GOTO 210
150 FUEL=(33.8461*LOA)-16.645+(N*(.012885+(.01538*LOA)))
160 JACKET=24.295-(3.4E+07/(N*N))-10*(1-LOA)
170 EXHAUST=((9.4614*LOA)-10.445+N*(.004845+(.01007*LOA)))*.8
180 QT1=QC+JACKET+EXHAUST:QT=QT1
190 IF QT1>=Q THEN Z=1
200 IF QT1>=Q GOTO 1620
210 REM *** DETERMINATION OF SCREW COMPRESSOR SERIES (R12) ***
220 IF ET2<-35 THEN LPRINT "Minimum Evaporating Temperature Too Low For Screw Co
mpressors":GOTO 3190
230 IF ET1>20 THEN LPRINT "Maximum Evaporating Temperature To High For Screw Com
pressors":GOTO 3190
240 ETMAX=-68.9922+9.8435*SQR(CT)
250 ETMIN=-60.6095+.539115*CT
260 IF ET2<ETMIN THEN LPRINT "Minimum Evaporating Temperature Too Low For Screw
Compressors":GOTO 3190
270 IF ET1<ETMAX THEN 1200
280 ETMAX=-65.1211+10.6651*SQR(CT)
290 ETMIN=-50.5097+.590079*CT
300 IF ET2<ETMIN THEN LPRINT "Evaporating Temperature Range Too Wide For Screw C
ompressors":GOTO 3190
310 IF ET1<ETMAX THEN 790
320 ETMAX=34.8785*LOG(CT)-108.71
330 ETMIN=-39+.65*CT
340 IF ET2<ETMIN THEN LPRINT "Evaporating Temperature Range Too Wide For Screw C
ompressors":GOTO 3190
350 IF ET1>ETMAX THEN LPRINT "Maximum Evaporating Temperature Too High For Screw
 Compressors":GOTO 3190
360 REM *** CALCULATIONS OF DESIGN DUTY FOR OSK(R12) COMPRESSORS ***
370 REM *** CALCULATIONS OF DESIGN DUTY FOR LEYLAND-BITZER(OSK-R12) UNIT  ***
380 QE=(79.4455*EXP(-1.38913E-04*CT*CT)+ET*(2.64398-2.03096E-04*CT*CT)+ET*ET*(.0
473872-9.33168E-04*CT+9.80597E-06*CT*CT))*N/2900
390 POWER=(-.233356+.446509*CT-1.03856E-03*CT*CT+ET*(-.240465+.0126049*CT-1.1935
8E-04*CT*CT)+ET*ET*(.0227649-8.75574E-04*CT+9.08092E
-06*CT*CT))*N/2900
400 QC=QE+POWER
410 LOA =POWER/(1/(.0215537+92694.6/(N*N)))
420 IF LOA > 1! GOTO 490
430 IF LOA < .4 GOTO 490
440 FUEL=(-17.5+.0475*N)*LOA
450 JACKET=(-12.5+.01875*N)*LOA
460 EXHAUST=(-10+.02*N)*LOA*.8
470 QT2=QC+JACKET+EXHAUST:QT=QT2
480 IF QT2>=Q GOTO 1820
490 REM *** CALCULATION OF DESIGN DUTY FOR SI4-BITZER(OSK-R12) UNIT ***
500 QE=(135.638*EXP(-1.36277E-04*CT*CT)+ET*(4.57923-3.7273E-04*CT*CT)+ET*ET*(.09
19746-2.07966E-03*CT+2.22842E-05*CT*CT))*N/2900
```

```
510 POWER=(3.43803+.564437*CT+ET*(-.531427+.0271301*CT-2.7165E-04*CT*CT)+ET*ET*(
.0412832-1.61896E-03*CT+1.70515E-05*CT*CT))*N/2900
520 QC=QE+POWER
530 LOA=POWER/((-327.771+52.1602* LOG(N))*.67)
540 IF LOA >1! THEN 610
550 IF LOA < .4 THEN 610
560 FUEL=.67*(34.2451+143.121*LOA-190.19*LOA*LOA-(2.30198E-03-.139836*LOA*LOA)*N
)
570 JACKET=.67*(29.4375-17.1687*LOA-(7.63154E-03-.0279542*LOA)*N)
580 EXHAUST=.67*(27.0529-54.0081*LOA*LOA-(4.00125E-03-.0695442*LOA*LOA)*N)*.8
590 QT3=QC+EXHAUST+JACKET:QT=QT3
600 IF QT3>=Q THEN 1860
610 REM *** CALCULATION OF DESIGN DUTY FOR SI6-BITZER(OSK-R12) UNIT ***
620 QE=(155.061*EXP(-1.36443E-04*CT*CT)+ET*(5.23143-4.25024E-04*(CT*CT))+ET*ET*(
.104687-2.36009E-03*CT+2.53098E-05*CT*CT))*N/2900
630 POWER=(1/(3.49025E-03+1.19659/CT)+ET*(-.592086+.0302465*CT-3.0121E-04*CT*CT)
+ET*ET*(.0473269-.0018571*CT+1.9441E-05*CT*CT))*N/29
00
640 QC=QE+POWER
650 LOA=POWER/(-327.771+52.1602*LOG(N))
660 IF LOA > 1! THEN PRINT "Engine Load Factor Too High":GOTO 3190
670 IF LOA < .4 THEN PRINT "Engine Load Factor Too Low":GOTO 3190
680 FUEL=34.2451+143.121*LOA-190.19*LOA*LOA-(2.30198E-03-.139836*LOA*LOA)*N
690 JACKET=29.4375-17.1687*LOA-(7.63154E-03-.0279542*LOA)*N
700 EXHAUST=27.0529-54.0081*LOA*LOA-(4.00125E-03-.0695442*LOA*LOA)*N*.8
710 QT4=QC+EXHAUST+JACKET:QT=QT4
720 IF QT>=Q THEN 1900
730 GOSUB 750
740 GOTO 780
750 IF COUNTER=1 GOTO 4790
760 LPRINT "The required performance is outside the range of this program"
770 RETURN
780 GOTO 3190
790 REM   *** CALCULATION OF DESIGN DUTY FOR OSN(R12) SERIES COMPRESSORS ***
800 REM   *** CALCULATION OF DESIGN DUTY FOR LEYLAND-BITZER(OSN-R12) MODULE ***
810 QE=(72.8605-6.46232E-03*CT*CT+ET*(1.94233+.0146278*CT-2.41207E-04*CT*CT)+ET*
ET*(.012758+2.74389E-04*CT-2.2328E-06*CT*CT))*N/2900
820 POWER=(14.022-.10474*CT+3.83115E-03*CT*CT+ET*(-.294964+.021014*CT-2.07748E-0
4*CT*CT)+ET*ET*(.0146469-5.43221E-04*CT+5.74369E-06*
CT*CT))*N/2900
830 QC=QE+POWER
840 LOA=POWER/(1/(.0215537+92694.6/(N*N)))
850 IF LOA>1 THEN 920
860 IF LOA<.4 THEN 920
870 FUEL=LOA*(-17.5+.0475*N)
880 JACKET=LOA*(.01875*N-12.5)
890 EXHAUST=LOA*(.02*N-10)*.8
900 QT2=QC+JACKET+EXHAUST:QT=QT2
910 IF QT>=Q THEN 3220
920 REM   *** CALCULATION OF DESIGN DUTY FOR SI4-BITZER(OSN-R12) MODULE ***
930 QE=(125.002-.0110863*CT*CT+ET*(3.33233+.0249833*CT-4.11486E-04*CT*CT)+ET*ET*
(.0219977+4.68202E-04*CT-3.86461E-06*CT*CT))*N/2900
940 POWER=(22.7326-.164887*CT+6.20828E-03*CT*CT+ET*(-.49839+.0348819*CT-3.43671E
-04*CT*CT)+ET*ET*(.0238289-8.86566E-04*CT+9.38828E-0
6*CT*CT))*N/2900
950 QC=QE+POWER
960 LOA=POWER/(.67*(-327.771+52.1602*LOG(N)))
970 IF LOA>1 THEN 1040
980 IF LOA<.4 THEN 1040
990 FUEL=.67*(34.2451+143.121*LOA-190.19*LOA*LOA-N*(2.30198E-03-.139836*LOA*LOA)
```

```
)
1000 JACKET=.67*(29.4375-17.1687*LOA-N*(7.63154E-03-.0279542*LOA))
1010 EXHAUST=.67*(27.0529-54.0081*LOA*LOA-N*(4.00125E-03-.0695442*LOA*LOA))*.8
1020 QT3=QC+EXHAUST+JACKET:QT=QT3
1030 IF QT3>Q THEN 3260
1040 REM   *** CALCULATION OF DESIGN DUTY FOR SI6-BITZER(DSN-R12) MODULE ***
1050 QE=N/2900*(142.865-.0126726*CT*CT+ET*(3.81371+.0283545*CT-4.68333E-04*CT*CT
)+ET*ET*(.0251286+5.37175E-04*CT-4.45133E-06*CT*CT))
1060 POWER=N/2900*(26.0021-.189453*CT+7.10531E-03*CT*CT+ET*(-.574547+.0400336*CT
-3.94488E-04*CT*CT)+ET*ET*(.0273648-1.02106E-03*CT+1
.08131E-05*CT*CT))
1070 QC=QE+POWER
1080 LOA=POWER/(-327.771+52.1602*LOG(N))
1090 IF LOA>1 THEN PRINT"Engine Load Factor Too High":GOTO 3190
1100 IF LOA<.4 THEN PRINT"Engine Load Factor Too Low":GOTO 3190
1110 FUEL=34.2451+143.121*LOA-190.19*LOA*LOA-N*(2.30198E-03-.139836*LOA*LOA)
1120 JACKET=29.4375-17.1687*LOA-N*(7.63154E-03-.0279542*LOA)
1130 EXHAUST=27.0529-54.0081*LOA*LOA-N*(4.00125E-03-.0695442*LOA*LOA)*.8
1140 QT4=QC+JACKET+EXHAUST:QT=QT4
1150 IF QT>=Q THEN 3300
1160 GOSUB 750
1170 GOTO 3190
1180 GOSUB 750
1190 GOTO 3190
1200 REM   *** CALCULATION OF DESIGN DUTY FOR DST(R12) SERIES COMPRESSORS ***
1210 REM   *** CALCULATION OF DESIGN DUTY FOR LEYLAND-BITZER(DST-R12) MODULE ***
1220 QE=N/2900*(69.8154-5.91753E-03*CT*CT+ET*(2.60467-.122425*SQR(CT))+ET*ET*(.0
28735-5.64313E-04*CT+5.02264E-06*CT*CT))
1230 POWER=N/2900*(15.6554-.12243*CT+3.97196E-03*CT*CT+ET*(.872213-.0239169*CT+2
.0562E-04*CT*CT)+ET*ET*(.0483475-1.47357E-03*CT+1.10
547E-05*CT*CT))
1240 QC=QE+POWER
1250 LOA=POWER/(1/(.0215537+92694.6 /(N*N)))
1260 IF LOA>1 THEN 1330
1270 IF LOA<.4 THEN 1330
1280 FUEL=LOA*(.0475*N-17.5)
1290 JACKET=LOA*(.01875*N-12.5)
1300 EXHAUST=LOA*(.02*N-10)*.8
1310 QT2=QC+EXHAUST+JACKET:QT=QT2
1320 IF QT>=Q THEN 3340
1330 REM   *** CALCULATION OF DESIGN DUTY FOR SI4-BITZER(DST-R12) MODULE ***
1340 QE=N/2900*(119.212-9.87824E-03*CT*CT+ET*1/(.251216+1.65676E-03*CT)+ET*ET*(.
020411+5.327/(CT*CT)))
1350 POWER=N/2900*(25.6827-.207076*CT+6.55666E-03*CT*CT+ET*(1.39842-.0383891*CT+
3.31614E-04*CT*CT)+ET*ET*(.0305224-1.02449E-03*CT+8.
46882E-06*CT*CT))
1360 QC=QE+POWER
1370 LOA=POWER/(.67*(-327.771+52.1602*LOG(N)))
1380 IF LOA>1 THEN 1450
1390 IF LOA<.4 THEN 1450
1400 FUEL=.67*(34.2451+143.121*LOA-190.19*LOA*LOA-N*(2.30198E-03-.139836*LOA*LOA
))
1410 JACKET=.67*(29.4375-17.1687*LOA-N*(7.63154E-03-.0279542*LOA))
1420 EXHAUST=.67*(27.0529-54.0081*LOA*LOA-N*(4.00125E-03-.0695442*LOA*LOA))*.8
1430 QT3=QC+JACKET+EXHAUST:QT=QT3
1440 IF QT3>=Q THEN 3380
1450 REM   *** CALCULATION OF DESIGN DUTY FOR SI6-BITZER(DST-R12) MODULE ***
1460 QE=N/2900*(136.202-.0112803*CT*CT+ET*(3.86411*EXP(-4.31348E-05*CT*CT))+ET*E
T*1/(42.2627-8.174/(CT*CT)))
1470 POWER=N/2900*(29.0072-.223498*CT+7.38276E-03*CT*CT+ET*(1.81382-.0511493*CT+
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4.39061E-04*CT*CT)+ET*ET*(.0355134-1.18916E-03*CT+9.
81644E-06*CT*CT))
1480 QC=QE+POWER
1490 LOA=POWER/(-327.771+52.1602*LOG(N))
1500 IF LOA>1 THEN PRINT"Engine Load Factor Too High":GOTO 3190
1510 IF LOA<.4 THEN PRINT"Engine Load Factor Too Low":GOTO 3190
1520 FUEL=34.2451+143.121*LOA-190.19*LOA*LOA-N*(2.30198E-03-.139836*LOA*LOA)
1530 JACKET=29.4375-17.1687*LOA-N*(7.63154E-03-.0279542*LOA)
1540 EXHAUST=27.0529-54.0081*LOA*LOA-N*(4.00125E-03-.0695442*LOA*LOA)*.8
1550 QT4=QC+JACKET+EXHAUST:QT=QT4
1560 IF QT>=Q THEN 3420
1570 GOSUB 750
1580 GOTO 3190
1590 GOSUB 750
1600 GOTO 3190
1610 REM    *** PERFORMANCE DATA BASED ON DESIGN CONDITIONS (R12) ***
1620 IF COUNTER=2 THEN GOSUB 4820
1630 LPRINT TAB(7) "Optimum Unit : FORD KENT - AGR 450 R12 Module "
1640 LPRINT "          ~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~ "
1650 LPRINT USING  "        TOTAL SYSTEM DUTY (kW) : £££.££";QT:LPRINT:LPRINT
1660 LPRINT "        Performance Characteristics"
1670 LPRINT "        ~~~~~~~~~~~~~~~~~~~~~~~~~~~"
1680 LPRINT "        Refrigerant Type .................... :  R12"
1690 LPRINT USING"        Evaporator Duty (kW) ................ : £££.££";QE
1700 LPRINT USING "        Power Absorbed (kW) ................. : £££.££";POWER
1710 LPRINT USING "        Condenser Duty (kW) ................. : £££.££";QC:LOA
=LOA*100
1720 LPRINT USING "        Engine Loading (%) .................. : £££.££";LOA
1730 LPRINT USING "        Fuel Usage (kW) ..................... : £££.££";FUEL
1740 LPRINT USING "        Exhaust Heat Recovery (kW) .......... : £££.££";EXHAUS
T
1750 LPRINT USING "        Jacket Water Heat Recovery (kW) ..... : £££.££";JACKET
1760 LPRINT USING "        Total System Duty (kW) .............. : £££.££";QT :PE
R=QT/FUEL
1770 LPRINT USING "        Primary Energy Ratio (P.E.R.) ....... : ££.££";PER :CO
P=QC/POWER
1780 LPRINT USING "        Coefficient Of Performance (C.O.P.) . : ££.££";COP :LP
RINT
1790 INPUT "Is Performance Over The Full Range Of Evaporating Temperature Requir
ed (YorN)?",A$
1800 IF A$="Y" THEN 1940
1810 IF A$<>"Y" THEN 3190
1820 IF COUNTER=2 THEN GOSUB 4820
1830 LPRINT TAB(7) "Optimum Unit : LEYLAND - BITZER 6061 OSK-R12 Module"
1840 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
1850 GOTO 1650
1860 IF COUNTER=2 THEN GOSUB 4820
1870 LPRINT TAB(7) "Optimum Unit : SI4 - BITZER 7051 OSK-R12 Module"
1880 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
1890 GOTO 1650
1900 IF COUNTER=2 THEN GOSUB 4820
1910 LPRINT TAB(7) "Optimum Unit : SI6 - BITZER 7061 OSK-R12 Module"
1920 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
1930 GOTO 1650
1940 REM *** CALCULATION OF UNIT DUTIES OVER A RANGE OF EVAP. TEMPS. (R12) ***
1950 REM *** FOR FORD-AGR AND BITZER OSK SERIES UNITS ***
1960 J=((ET1-ET2)\5)+1
1970 EVAP(1)=ET2
1980 IF Q<=Q11 THEN 2600
1990 IF Q<=Q12 THEN 2400
```

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2000 IF Q<=QT3 THEN 2200
2010 REM   *** SI6-BITZER(DSK) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R12) ***
2020 FOR I=1 TO J
2030 QE(I)=(155.061*EXP(-1.36443E-04*CT*CT)+EVAP(I)*(5.23143-4.25024E-04*(CT*CT)
)+EVAP(I)*EVAP(I)*(.104687-2.36009E-03*CT+2.53098E-0
5*CT*CT))*N/2900
2040 POWER(I)=(1/(3.49025E-03+1.19659/CT)+EVAP(I)*(-.592086+.0302465*CT-3.0121E-
04*CT*CT)+EVAP(I)*EVAP(I)*(.0473269-.0018517*CT+1.94
41E-05*CT*CT))*N/2900
2050 QC(I)=QE(I)+POWER(I):LOA(I)=POWER(I)/(-327.771+52.1602*LOG(N))
2060 IF LOA(I)<.4 THEN 2160
2070 IF LOA(I)>1! THEN 2160
2080 FUEL(I)=34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)+N*(-2.30198E-03+.139836
*LOA(I)*LOA(I))
2090 JACKET(I)=29.4375-17.1687*LOA(I)+N*(-7.63154E-03+.0279542*LOA(I))
2100 EXHAUST(I)=27.0529-54.0081*LOA(I)*LOA(I)+N*(-4.00125E-03+.0695442*LOA(I)*LO
A(I))*.8
2110 HEATREC(I)=JACKET(I)+EXHAUST(I)
2120 QT(I)=QC(I)+HEATREC(I)
2130 PER(I)=QT(I)/FUEL(I)
2140 COP(I)=QC(I)/POWER(I)
2150 GOTO 2170
2160 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
2170 EVAP(I+1)=EVAP(I)+5
2180 NEXT I
2190 GOTO 2820
2200 REM *** SI4-BITZER(DSK) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R12) ***
2210 FOR I=1 TO J
2220 QE(I)=(135.638*EXP(-1.36272E-04*CT*CT)+EVAP(I)*(4.57923-3.7273E-04*CT*CT)+E
VAP(I)*EVAP(I)*(.0919746-2.07966E-03*CT+2.22842E-05*
CT*CT))*N/2900
2230 POWER(I)=(3.43803+.564437*CT+EVAP(I)*(-.531427+.0271301*CT-2.7165E-04*CT*CT
)+EVAP(I)*EVAP(I)*(.0412832-1.61896E-03*CT+1.70515E-
05*CT*CT))*N/2900
2240 QC(I)=QE(I)+POWER(I)
2250 LOA(I)=POWER(I)/(.67*(-327.771+52.1602*LOG(N)))
2260 IF LOA(I)<.4 THEN 1990
2270 IF LOA(I)>1 THEN 1990
2280 FUEL(I)=.67*(34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)-N*(2.30198E-03-.13
9836*LOA(I)*LOA(I)))
2290 JACKET(I)=.67*(29.4375-17.1687*LOA(I)-N*(7.63154E-03-.0279542*LOA(I)))
2300 EXHAUST(I)=.67*(27.0529-54.0081*LOA(I)*LOA(I)-N*(4.00125E-03-.0695442*LOA(I
)*LOA(I)))*.8
2310 QT(I)=QC(I)+JACKET(I)+EXHAUST(I)
2320 HEATREC(I)=EXHAUST(I)+JACKET(I)
2330 PER(I)=QT(I)/FUEL(I)
2340 COP(I)=QC(I)/POWER(I)
2350 GOTO 2370
2360 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
2370 EVAP(I+1)=EVAP(I)+5
2380 NEXT I
2390 GOTO 2850
2400 REM *** LEYLAND-BITZER(DSK) PERFORMANCE OVER RANGE OF EVAP. TEMP.(R12) ***
2410 FOR I=1 TO J
2420 QE(I)=(79.4455*EXP(-1.38913E-04*CT*CT)+EVAP(I)*(2.64398-2.03096E-04*CT*CT)+
EVAP(I)*EVAP(I)*(.0473872-9.33168E-04*CT+9.80597E-06
*CT*CT))*N/2900
2430 POWER(I)=(-.233356+.446509*CT-1.03856E-03*CT*CT+EVAP(I)*(-.240465+.0126049*
```

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CT-1.19358E-04*CT*CT)+EVAP(I)*EVAP(I)*(.0227649-8.75
574E-04*CT+9.08092E-06*CT*CT))*N/2900
2440 OC(I)=OE(I)+POWER(I)
2450 LOA(I)=POWER(I)/(1/(.0215537+92694.6/(N*N)))
2460 IF LOA(I)<.4 THEN 2560
2470 IF LOA(I)>1 THEN 2560
2480 FUEL(I)=LOA(I)*(-17.5+.0475*N)
2490 JACKET(I)=LOA(I)*(-12.5+.01875*N)
2500 EXHAUST(I)=LOA(I)*(.02*N-10)*.8
2510 HEATREC(I)=EXHAUST(I)+JACKET(I)
2520 OT(I)=OC(I)+HEATREC(I)
2530 PER(I)=OT(I)/FUEL(I)
2540 COP(I)=OC(I)/POWER(I)
2550 GOTO 2570
2560 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:OT(I)=0:PER(I)=0:COP(I)=0:OC(I)=0:HEATRE
C(I)=0
2570 EVAP(I+1)=EVAP(I)+5
2580 NEXT I
2590 GOTO 2880
2600 REM *** FORD-AGR PERFORMANCE OVER RANGE OF EVAP.TEMP.(R12) ***
2610 FOR I=1 TO J
2620 ETF(I)=EVAP(I)*1.8+32
2630 OE(I)=(N/3000)*(37.4243+(.96648*ETF(I))+(.002298*ETF(I)*ETF(I))+((-.17383+(
.003237*ETF(I)))*CTF))
2640 POWER(I)=(N/3000)*(10.025*EXP((.002206+.0000256*ETF(I))*CTF))/1.0526
2650 OC(I)=OE(I)+POWER(I)
2660 LOA(I)=(-.0006823+108.804/N)*POWER(I)^(1/(.9872+1203501/(N*N)))
2670 IF LOA(I)<.4 THEN 2770
2680 IF LOA(I)>1 THEN 2770
2690 FUEL(I)=(33.8461*LOA(I))-16.654+(N*(.012885+(.01538*LOA(I))))
2700 JACKET(I)=24.295-(3.4E+07/(N*N))-10*(1-LOA(I))
2710 EXHAUST(I)=((9.4614*LOA(I))-10.445+N*(.004845+(.01007*LOA(I))))*.8
2720 HEATREC(I)=JACKET(I)+EXHAUST(I)
2730 OT(I)=OC(I)+HEATREC(I)
2740 PER(I)=OT(I)/FUEL(I)
2750 COP(I)=OC(I)/POWER(I)
2760 GOTO 2780
2770 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:OT(I)=0:PER(I)=0:COP(I)=0:OC(I)=0:HEATRE
C(I)=0
2780 EVAP(I+1)=EVAP(I)+5
2790 NEXT I
2800 GOTO 2910
2810 REM    *** TABULATED PERFORMANCE DATA ***
2820 LPRINT TAB(7) "Tabulated Data For B16-BitzerOSK Module :"
2830 LPRINT TAB(7) "*****************************************";LPRINT
2840 GOTO 3050
2850 LPRINT TAB(7) "Tabulated Data For B14-BitzerOSK Module :"
2860 LPRINT TAB(7) "*****************************************";LPRINT
2870 GOTO 3050
2880 LPRINT TAB(7)"Tabulated Data For Leyland-BitzerOSK Module :"
2890 LPRINT TAB(7)"**********************************************";LPRINT
2900 GOTO 3050
2910 LPRINT TAB(7)"Tabulated Data For Ford-AGR Module :"
2920 LPRINT TAB(7)"************************************";LPRINT:GOTO 3050
2930 LPRINT TAB (7) "TABULATED DATA FOR B16-BITZER(OSN) MODULE "
2940 LPRINT TAB(7) "****************************************** ";LPRINT:GOTO 3050
2950 LPRINT TAB(7) "TABULATED DATA FOR B14-BITZER(OSN) MODULE "
2960 LPRINT TAB(7) "****************************************** ";LPRINT:GOTO 3050
2970 LPRINT TAB(7) "TABULATED DATA FOR LEYLAND-BITZER(OSN) MODULE "
2980 LPRINT TAB(7) "********************************************** ";LPRINT:GOTO
```

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3050
2990 LPRINT TAB(7) "TABULATED DATA FOR SI6-BITZER(OST) MODULE "
3000 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~ ":LPRINT:GOTO 3050
3010 LPRINT TAB(7) "TABULATED DATA FOR SI4-BITZER(OST) MODULE "
3020 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~ ":LPRINT:GOTO 3050
3030 LPRINT TAB(7) "TABULATED DATA FOR LEYLAND-BITZER(OST) MODULE "
3040 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~ ":LPRINT:GOTO
3050
3050 LPRINT "       Refrigerant Type ................... :  R12 "
3060 LPRINT"       Condensing Temperature (Deg C) ...... : ";CT:LPRINT
3070 LPRINT"          Evap   Power    Eng    Condr   Heat    Total   Fuel     CO
P    PER"
3080 LPRINT "           Temp    Abs      Load    Duty     Rec     Duty    Usage"
3090 LPRINT "           (C)    (kW)             (kW)    (kW)     (kW)    (kW)"
3100 LPRINT
3110 FOR I=1 TO J
3120 LPRINT USING "          £££.£   ££.£     £.££   £££.£   £££.£   £££.£   £££.
£    £.££   £.££";EVAP(I);POWER(I);LOA(I);DC(I);HEAT
REC(I);QT(I);FUEL(I);COP(I);PER(I)
3130 NEXT I
3140 IF Z=1 THEN 3190
3150 LPRINT:LPRINT:LPRINT
3160 LPRINT TAB(7) "Limits of operation ":LPRINT
3170 LPRINT USING  "        Max. evaporating temperature deg C : £££.££";ETMAX
3180 LPRINT USING  "        Min. evaporating temperature deg C : £££.££";ETMIN
3190 INPUT "Are Any Other Checks Required (YorN)? :",A$
3200 IF A$="Y" THEN PRINT "RETURNING TO OPERATING PROGRAM":LOAD "GEHP-22",R
3210 IF A$<>"Y" THEN 4840
3220 IF COUNTER=2 THEN GOSUB 4820
3230 LPRINT TAB(7) "Optimum Unit : LEYLAND - BITZER 6061 OSN-R12  Module "
3240 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3250 GOTO 1650
3260 IF COUNTER=2 THEN GOSUB 4820
3270 LPRINT TAB(7) "Optimum Unit :SI4 - BITZER 7051 OSN-R12 Module"
3280 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3290 GOTO 1650
3300 IF COUNTER=2 THEN GOSUB 4820
3310 LPRINT TAB(7) "Optimum Unit :SI6 - BITZER 7061 OSN-R12 Module"
3320 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3330 GOTO 1650
3340 IF COUNTER=2 THEN GOSUB 4820
3350 LPRINT TAB(7) "Optimum Unit : LEYLAND - BITZER 6061 OST-R12 Module"
3360 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3370 GOTO 1650
3380 IF COUNTER=2 THEN GOSUB 4820
3390 LPRINT TAB(7) "Optimum Unit : SI4 - BITZER 7051 OST-R12 Module"
3400 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3410 GOTO 1650
3420 IF COUNTER=2 THEN GOSUB 4820
3430 LPRINT TAB(7) "Optimum Unit : SI6 - BITZER 7061 OST-R12 Module"
3440 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
3450 GOTO 1650
3460 REM *** CALCULATION OF UNIT DUTIES OVER A RANGE OF EVAP. TEMPS. (R12) ***
3470 REM *** FOR BITZER OSN SERIES UNITS ***
3480 J=((ET1-ET2)\5)+1
3490 EVAP(1)=ET2
3500 IF Q<=QT2 THEN 3910
3510 IF Q<=QT3 THEN 3710
3520 REM   *** SI6-BITZER(OSN) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R12) ***
3530 FOR I= 1 TO J
```

```
3540 QE(I)=(142.865-.0126726*CT*CT+EVAP(I)*(3.81371+.0283545*CT-4.68333E-04*CT*C
T)+EVAP(I)*EVAP(I)*(.0251286+5.37175E-04*CT-4.45133E
-06*CT*CT))*N/2900
3550 POWER(I)=(26.0021-.189453*CT+7.10531E-03*CT*CT+EVAP(I)*(-.574547+.0400336*C
T-3.94488E-04*CT*CT)+EVAP(I)*EVAP(I)*(.0273648-1.021
06E-03*CT+1.08131E-05*CT*CT))*N/2900
3560 QC(I)=QE(I)+POWER(I):LOA(I)=POWER(I)/(-327.771+52.1602*LOG(N))
3570 IF LOA(I)<.4 THEN 3670
3580 IF LOA(I)>1! THEN 3670
3590 FUEL(I)=34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)+N*(-2.30198E-03+.139836
*LOA(I)*LOA(I))
3600 JACKET(I)=29.4375-17.1687*LOA(I)+N*(-7.63154E-03+.0279542*LOA(I))
3610 EXHAUST(I)=27.0529-54.0081*LOA(I)*LOA(I)+N*(-4.00125E-03+.0695442*LOA(I)*LO
A(I))*.8
3620 HEATREC(I)=JACKET(I)+EXHAUST(I)
3630 QT(I)=QC(I)+HEATREC(I)
3640 PER(I)=QT(I)/FUEL(I)
3650 COP(I)=QC(I)/POWER(I)
3660 GOTO 3680
3670 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
3680 EVAP(I+1)=EVAP(I)+5
3690 NEXT I
3700 GOTO 2930
3710 REM *** SI4-BITZER(DSN) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R12) ***
3720 FOR I=1 TO J
3730 QE(I)=(125.002-.0110863*CT*CT+EVAP(I)*(3.33233+.0249833*CT-4.11486E-04*CT*C
T)+EVAP(I)*EVAP(I)*(.0219977+4.68202E-04*CT-3.86461E
-06*CT*CT))*N/2900
3740 POWER(I)=(22.7326-.164887*CT+6.20828E-03*CT*CT+EVAP(I)*(-.49839+.0348819*CT
-3.43671E-04*CT*CT)+EVAP(I)*EVAP(I)*(.0238289-8.8656
6E-04*CT+9.38828E-06*CT*CT))*N/2900
3750 QC(I)=QE(I)+POWER(I)
3760 LOA(I)=POWER(I)/(.67*(-327.771+52.1602*LOG(N)))
3770 IF LOA(I)<.4 THEN 3870
3780 IF LOA(I)>1 THEN 3870
3790 FUEL(I)=.67*(34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)-N*(2.30198E-03-.13
9836*LOA(I)*LOA(I)))
3800 JACKET(I)=.67*(29.4375-17.1687*LOA(I)-N*(7.63154E-03-.0279542*LOA(I)))
3810 EXHAUST(I)=.67*(27.0529-54.0081*LOA(I)*LOA(I)-N*(4.00125E-03-.0695442*LOA(I
)*LOA(I)))*.8
3820 QT(I)=QC(I)+JACKET(I)+EXHAUST(I)
3830 HEATREC(I)=JACKET(I)+EXHAUST(I)
3840 PER(I)=QT(I)/FUEL(I)
3850 COP(I)=QC(I)/POWER(I)
3860 GOTO 3880
3870 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
3880 EVAP(I+1)=EVAP(I)+5
3890 NEXT I
3900 GOTO 2950
3910 REM *** LEYLAND-BITZER(DSN) PERFORMANCE OVER RANGE OF EVAP. TEMP.(R12) ***
3920 FOR I=1 TO J
3930 QE(I)=(72.8605-6.46232E-03*CT*CT+EVAP(I)*(1.94233+.0146278*CT-2.41207E-04*C
T*CT)+EVAP(I)*EVAP(I)*(.012758+2.74389E-04*CT-2.2328
E-06*CT*CT))*N/2900
3940 POWER(I)=(14.022-.10474*CT+3.83115E-03*CT*CT+EVAP(I)*(-.294964+.021014*CT-2
.07748E-04*CT*CT)+EVAP(I)*EVAP(I)*(.0146469-5.43221E
-04*CT+5.74369E-06*CT*CT))*N/2900
3950 QC(I)=QE(I)+POWER(I)
```

```
3960 LOA(I)=POWER(I)/(1/(.0215537+92694.6/(N*N)))
3970 IF LOA(I)<.4 THEN 4070
3980 IF LOA(I)>1 THEN 4070
3990 FUEL(I)=LOA(I)*(-17.5+.0475*N)
4000 JACKET(I)=LOA(I)*(-12.5+.01875*N)
4010 EXHAUST(I)=LOA(I)*(.02*N-10)*.8
4020 HEATREC(I)=EXHAUST(I)+JACKET(I)
4030 QT(I)=QC(I)+HEATREC(I)
4040 PER(I)=QT(I)/FUEL(I)
4050 COP(I)=QC(I)/POWER(I)
4060 GOTO 4080
4070 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
4080 EVAP(I+1)=EVAP(I)+5
4090 NEXT I
4100 GOTO 2970
4110 REM *** CALCULATION OF UNIT DUTIES OVER A RANGE OF EVAP. TEMPS. (R12) ***
4120 REM *** FOR BITZER OST SERIES UNITS ***
4130 J=((ET1-ET2)\5)+1
4140 EVAP(1)=ET2
4150 IF Q<=QT2 THEN 4580
4160 IF Q<=QT3 THEN 4370
4170 REM    *** S16-BITZER(OST) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R12) ***
4180 FOR I=1 TO J
4190 QE(I)=(136.202-.0112803*CT*CT+EVAP(I)*(3.86411*EXP(-4.31348E-05*CT*CT))+EVA
P(I)*EVAP(I)/(42.2627-8174.12/(CT*CT)))*N/2900
4200 POWER(I)=29.0072-.223498*CT+7.38276E-03*CT*CT+EVAP(I)*(1.81382-.0511493*CT+
4.39061E-04*CT*CT)+EVAP(I)*EVAP(I)*(.0355134-1.18916
E-03*CT+9.81644E-06*CT*CT)
4210 POWER(I)=POWER(I)*N/2900
4220 QC(I)=QE(I)+POWER(I):LOA(I)=POWER(I)/(-327.771+52.1602*LOG(N))
4230 IF LOA(I)<.4 THEN 4330
4240 IF LOA(I)>1 THEN 4330
4250 FUEL(I)=34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)+N*(-2.30198E-03+.139836
*LOA(I)*LOA(I))
4260 JACKET(I)=29.4375-17.1687*LOA(I)+N*(-7.63154E-03+.0279542*LOA(I))
4270 EXHAUST(I)=27.0529-54.0081*LOA(I)*LOA(I)+N*(-4.00125E-03+.0695442*LOA(I)*LO
A(I))*.8
4280 HEATREC(I)=JACKET(I)+EXHAUST(I)
4290 QT(I)=QC(I)+HEATREC(I)
4300 PER(I)=QT(I)/FUEL(I)
4310 COP(I)=QC(I)/POWER(I)
4320 GOTO 4340
4330 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
4340 EVAP(I+1)=EVAP(I)+5
4350 NEXT I
4360 GOTO 2990
4370 REM *** SI4-BITZER(OST) PERFORMANCE OVER RANGE OF EVAP. TEMP. (R12) ***
4380 FOR I=1 TO J
4390 QE(I)=(119.212-9.87824E-03*CT*CT+EVAP(I)/(.251216+1.65676E-03*CT)+EVAP(I)*E
VAP(I)*(.020411+5.327/(CT*CT)))*N/2900
4400 POWER(I)=25.6827-.207076*CT+6.55666E-03*CT*CT+EVAP(I)*(1.39842-.0383891*CT+
3.31614E-04*CT*CT)+EVAP(I)*EVAP(I)*(.0305224-1.02449
E-03*CT+8.46882E-06*CT*CT)
4410 POWER(I)=POWER(I)*N/2900
4420 QC(I)=QE(I)+POWER(I)
4430 LOA(I)=POWER(I)/(.67*(-327.771+52.1602*LOG(N)))
4440 IF LOA(I)<.4 THEN 4540
4450 IF LOA(I)>1 THEN 4540
```

```
4460 FUEL(I)=.67*(34.2451+143.121*LOA(I)-190.19*LOA(I)*LOA(I)-N*(2.30198E-03-.13
9836*LOA(I)*LOA(I)))
4470 JACKET(I)=.67*(29.4375-17.1687*LOA(I)-N*(7.63154E-03-.0279542*LOA(I)))
4480 EXHAUST(I)=.67*(27.0529-54.0081*LOA(I)*LOA(I)-N*(4.00125E-03-.0695442*LOA(I
)*LOA(I)))*.8
4490 HEATREC(I)=JACKET(I)+EXHAUST(I)
4500 QT(I)=QC(I)+JACKET(I)+EXHAUST(I)
4510 PER(I)=QT(I)/FUEL(I)
4520 COP(I)=QC(I)/POWER(I)
4530 GOTO 4550
4540 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
4550 EVAP(I+1)=EVAP(I)+5
4560 NEXT I
4570 GOTO 3010
4580 REM *** LEYLAND-BITZER(OST) PERFORMANCE OVER RANGE OF EVAP. TEMP.(R12) ***
4590 FOR I=1 TO J
4600 QE(I)=(69.8154-5.91753E-03*CT*CT+EVAP(I)*(2.60467-.122425*SQR(CT))+EVAP(I)*
EVAP(I)*(.028735-5.64313E-04*CT+5.02264E-06*CT*CT))*
N/2900
4610 POWER(I)=15.6554-.12243*CT+3.97196E-03*CT*CT+EVAP(I)*(.872213-.0239169*CT+2
.0562E-04*CT*CT)+EVAP(I)*EVAP(I)*(.0483475-1.47357E-
03*CT+1.10547E-05*CT*CT)
4620 POWER(I)=POWER(I)*N/2900
4630 QC(I)=QE(I)+POWER(I)
4640 LOA(I)=POWER(I)/(1/(.0215537+92694.6/(N*N)))
4650 IF LOA(I)<.4 THEN 4750
4660 IF LOA(I)>1 THEN 4750
4670 FUEL(I)=LOA(I)*(-17.5+.0475*N)
4680 JACKET(I)=LOA(I)*(-12.5+.01875*N)
4690 EXHAUST(I)=LOA(I)*(.02*N-10)*.8
4700 HEATREC(I)=EXHAUST(I)+JACKET(I)
4710 QT(I)=QC(I)+HEATREC(I)
4720 PER(I)=QT(I)/FUEL(I)
4730 COP(I)=QC(I)/POWER(I)
4740 GOTO 4760
4750 FUEL(I)=0:JACKET(I)=0:EXHAUST(I)=0:QT(I)=0:PER(I)=0:COP(I)=0:QC(I)=0:HEATRE
C(I)=0
4760 EVAP(I+1)=EVAP(I)+5
4770 NEXT I
4780 GOTO 3030
4790 Q=Q/2
4800 COUNTER=2
4810 GOTO 60
4820 LPRINT TAB(7) "TWO MODULES REQUIRED : PERFORMANCE OF A SINGLE MODULE FOLLOW
S:":LPRINT
4830 RETURN
4840 END
```

C) TYPICAL PRINTOUTS FROM THE G.E.H.P. COMPUTER DESIGN AID PROGRAMS

GEHP SELECTION PROGRAM

Design Conditions :

```
Evaporating Temperature ... Deg C : -10
Condensing Temperature .... Deg C :  70
Heat Load Requirement ..... kW    :  180
Operating Speed ........... rpm   :  2600
```

```
Anticipated range of evaporating temperature
Maximum evaporating temperature ....deg C :  10
Minimum evaporating temperature ....deg C : -10
```

TWO MODULES REQUIRED : PERFORMANCE OF A SINGLE MODULE FOLLOWS:

Optimum Unit : LEYLAND - BITZER 6061 DST-R12 Module

TOTAL SYSTEM DUTY (kW) : 99.52

Performance Characteristics

```
Refrigerant Type ..................... :  R12
Evaporator Duty (kW) ................. :  23.67
Power Absorbed (kW) .................. :  21.90
Condenser Duty (kW) .................. :  45.57
Engine Loading (%) ................... :  77.24
Fuel Usage (kW) ...................... :  81.87
Exhaust Heat Recovery (kW) ........... :  25.95
Jacket Water Heat Recovery (kW) ...... :  28.00
Total System Duty (kW) ............... :  99.52
Primary Energy Ratio (P.E.R.) ........ :  1.22
Coefficient Of Performance (C.O.P.) .. :  2.08
```

Tabulated Data For Leyland-BitzerDSK Module :

```
Refrigerant Type ..................... :  R12
Condensing Temperature (Deg C) ....... :  70
```

Evap Temp (C)	Power Abs (kW)	Eng Load	Condr Duty (kW)	Heat Rec (kW)	Total Duty (kW)	Fuel Usage (kW)	COP	PER
-10.0	23.3	0.82	47.3	57.3	104.6	87.0	2.03	1.20
-5.0	23.1	0.82	52.5	57.0	109.4	86.5	2.27	1.27
0.0	23.3	0.82	59.3	57.3	116.6	86.9	2.55	1.34
5.0	23.6	0.83	67.8	58.2	126.0	88.4	2.87	1.43
10.0	24.3	0.86	77.8	59.9	137.7	90.8	3.20	1.52

Limits of operation

```
Max. evaporating temperature deg C :  13.36
Min. evaporating temperature deg C : -22.87
```

GEHP SELECTION PROGRAM

Design Conditions :

Evaporating Temperature ... Deg C : -10
Condensing Temperature .... Deg C : 45
Heat Load Requirement ..... kW : 90
Operating Speed .... rpm : 2600

Anticipated range of evaporating temperature
Maximum evaporating temperature ....deg C : 10
Minimum evaporating temperature ....deg C : -10

Optimum Unit : FORD KENT-AGR450 R22 Module

TOTAL SYSTEM DUTY (kW) : 95.77

Performance Characteristics

Refrigerant Type	:	R22
Evaporator Duty (kW)	:	34.52
Power Absorbed (kW)	:	19.62
Condenser Duty (kW)	:	54.14
Engine Loading (%)	:	79.57
Fuel Usage (kW)	:	75.60
Exhaust Heat Recovery (kW)	:	24.41
Jacket Water Heat Recovery (kW)	:	17.22
Total System Duty (kW)	:	95.77
Primary Energy Ratio (P.E.R.)	:	1.27
Coefficient Of Performance (C.O.P.) .	:	2.76

Tabulated Data For Ford-AGR Module :

Refrigerant Type .... : R22
Condensing Temperature (Deg C) ...... : 45

Evap Temp (C)	Power Abs (kW)	Eng Load	Condr Duty (kW)	Heat Rec (kW)	Total Duty (kW)	Fuel Usage (kW)	COP	PER
-10.0	19.6	0.80	54.1	41.6	95.8	75.6	2.76	1.27
-5.0	20.4	0.83	63.3	42.9	106.1	78.0	3.10	1.36
0.0	21.4	0.87	73.4	44.4	117.8	81.0	3.43	1.45
5.0	22.6	0.92	84.4	46.3	130.7	84.6	3.73	1.55
10.0	24.1	0.98	96.4	48.5	145.0	88.8	4.01	1.63

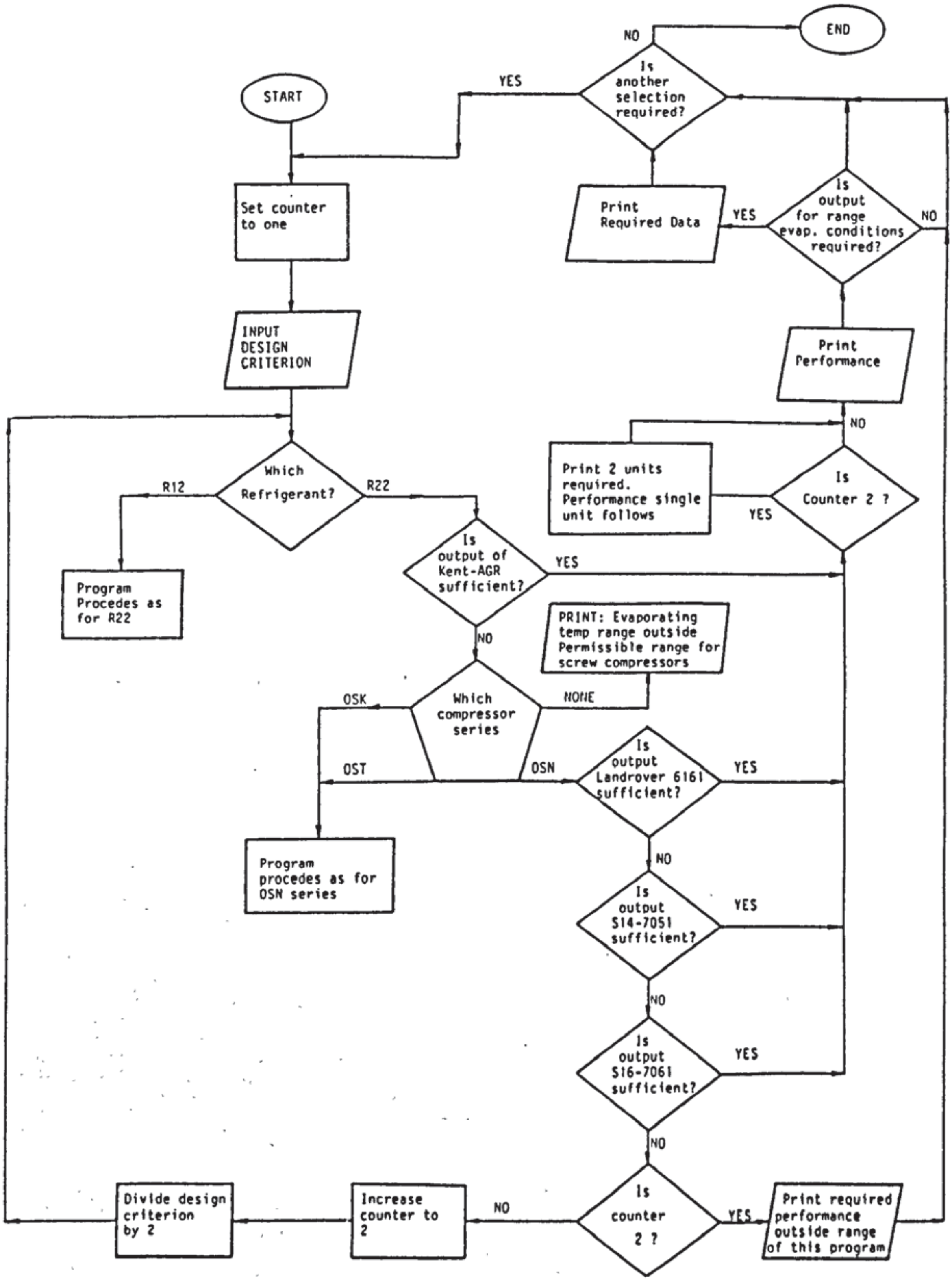

Figure A4.4.1 Flow Chart For G.E.H.P. Computer Design Aid

APPENDIX 4.5

PROGRAM LISTING: CURVE FITTING PROGRAM

```
10 REM   *** BILL BROWN  13/10/1982 ***
20 REM   *** PROGRAM NAME  : CURVFIT ***
30 PRINT:PRINT:PRINT
40 PRINT "CURVE FITTING PROGRAMME ."
50 PRINT "~~~~~~~~~~~~~~~~~~~~~~~~~~~"
60 PRINT "If you require a PRINTOUT ensure that the printer is switched ON ,"
70 PRINT "if not you may switch the printer OFF . If printer is already off"
80 PRINT "and you require a PRINTOUT then you need to run programme FORM first .
"
90 LPRINT TAB(7) "CURVE FITTING PROGRAMME ."
100 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~":LPRINT
110 PRINT:INPUT "Number of Points to be entered ? ",N:PRINT
120 DIM X(N),Y(N)
130 FOR T=1 TO N
140 INPUT "Value of X ...... : ";X(T)
150 LPRINT TAB(14) "Value of X ......... : ";X(T)
160 INPUT "Value of Y ...... : ";Y(T):PRINT
170 LPRINT TAB(14) "Value of Y ......... : ";Y(T):LPRINT
180 NEXT T
190 PRINT "*** LAW ***              ** R^2 **   *** A ***      *** B ***    *** C
 ***"
200 LPRINT "*** LAW ***" TAB(23) "* R^2 *" TAB(38) "*** A ***" TAB(50) "*** B **
*" TAB(66) "*** C ***"
210 PRINT "-------------------------------------------------------------------
--"
220 LPRINT "------------------------------------------------------------------
-----"
230 F=0:G=0:H=0:J=0:K=0:L=0:M=0
240 L$=" 1 : Y = A+BX"
250 FOR Z=1 TO N
260 C=X(Z):D=Y(Z):L=X(Z):M=Y(Z)
270 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
280 NEXT Z
290 R1=1
300 GOSUB 2010
310 R21=A:R22=B
320 F=0:G=0:H=0:J=0:K=0:L=0:M=0
330 L$=" 2 : Y = 1/(A+BX)"
340 FOR Z=1 TO N
350 L=X(Z):M=Y(Z)
360 IF M=0 THEN R2=0:GOTO 440
370 IF M<>0 GOTO 380
380 C=X(Z):D=1/Y(Z)
390 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
400 NEXT Z
410 R2=1
420 GOSUB 2010
430 R23=A:R24=B
440 F=0:G=0:H=0:J=0:K=0:L=0:M=0
450 L$=" 3 : Y = A+B/X"
460 FOR Z=1 TO N
470 C=1/X(Z):D=Y(Z):L=X(Z):M=Y(Z)
480 IF L=0 THEN R3=0:GOTO 550
490 IF L<>0 GOTO 500
500 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
510 NEXT Z
520 R3=1
530 GOSUB 2010
540 R25=A:R26=B
550 F=0:G=0:H=0:J=0:K=0:L=0:M=0
```

```
560 L$=" 4 : Y = 1/(A+B/X)"
570 FOR Z=1 TO N
580 L=X(Z):M=Y(Z)
590 IF L=0 THEN R4=0:GOTO 690
600 IF L<>0 GOTO 610
610 IF M=0 THEN R4=0:GOTO 690
620 IF M<>0 GOTO 630
630 C=1/X(Z):D=1/Y(Z)
640 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
650 NEXT Z
660 R4=1
670 GOSUB 2010
680 R27=A:R28=B
690 F=0:G=0:H=0:J=0:K=0:L=0:M=0
700 L$=" 5 : Y = A+BX^2"
710 FOR Z=1 TO N
720 C=X(Z)*X(Z):D=Y(Z):L=X(Z):M=Y(Z)
730 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
740 NEXT Z
750 R5=1
760 GOSUB 2010
770 R29=A:R30=B
780 F=0:G=0:H=0:J=0:K=0:L=0:M=0
790 L$=" 6 : Y = A+B/X^2"
800 FOR Z=1 TO N
810 L=X(Z):M=Y(Z)
820 IF L=0 THEN R6=0:GOTO 900
830 IF L<>0 GOTO 840
840 C=1/(X(Z)*X(Z)):D=Y(Z)
850 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
860 NEXT Z
870 R6=1
880 GOSUB 2010
890 R31=A:R32=B
900 F=0:G=0:H=0:J=0:K=0:L=0:M=0
910 L$=" 7 : Y = 1/(A+B/X^2)"
920 FOR Z=1 TO N
930 L=X(Z):M=Y(Z)
940 IF L=0 THEN R7=0:GOTO 1040
950 IF L<>0 GOTO 960
960 IF M=0 THEN R7=0:GOTO 1040
970 IF M<>0 GOTO 980
980 C=1/(X(Z)*X(Z)):D=1/Y(Z)
990 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
1000 NEXT Z
1010 R7=1
1020 GOSUB 2010
1030 R33=A:R34=B
1040 F=0:G=0:H=0:J=0:K=0:L=0:M=0
1050 L$=" 8 : Y = A+B*SQR(X)"
1060 FOR Z=1 TO N
1070 L=X(Z):M=Y(Z)
1080 IF L<0 THEN R8=0:GOTO 1160
1090 IF L=>0 GOTO 1100
1100 C=SQR(X(Z)):D=Y(Z)
1110 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
1120 NEXT Z
1130 R8=1
1140 GOSUB 2010
1150 R35=A:R36=B
```

```
1160 F=0:G=0:H=0:J=0:K=0:L=0:M=0
1170 L$=" 9 : Y = A+B/SQR(X)"
1180 FOR Z=1 TO N
1190 L=X(Z):M=Y(Z)
1200 IF L<=0 THEN R9=0:GOTO 1280
1210 IF L>0 GOTO 1220
1220 C=1/SQR(X(Z)):D=Y(Z)
1230 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
1240 NEXT Z
1250 R9=1
1260 GOSUB 2010
1270 R37=A:R38=B
1280 F=0:G=0:H=0:J=0:K=0:L=0:M=0
1290 L$="10 : Y = 1/(A+B/SQR(X))"
1300 FOR Z=1 TO N
1310 L=X(Z):M=Y(Z)
1320 IF L<=0 THEN R10=0:GOTO 1420
1330 IF L>0 GOTO 1340
1340 IF M=0 THEN R10=0:GOTO 1420
1350 IF M<>0 GOTO 1360
1360 C=1/(SQR(X(Z))):D=1/Y(Z)
1370 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
1380 NEXT Z
1390 R10=1
1400 GOSUB 2010
1410 R39=A:R40=B
1420 F=0:G=0:H=0:J=0:K=0:L=0:M=0
1430 L$="11 : Y = A*X^B"
1440 FOR Z=1 TO N
1450 L=X(Z):M=Y(Z)
1460 IF L<=0 THEN R11=0:GOTO 1560
1470 IF L>0 GOTO 1480
1480 IF M<=0 THEN R11=0:GOTO 1560
1490 IF M>0 GOTO 1500
1500 C=LOG(X(Z)):D=LOG(Y(Z))
1510 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
1520 NEXT Z
1530 R11=1
1540 GOSUB 2080
1550 R41=A:R42=B
1560 F=0:G=0:H=0:J=0:K=0:L=0:M=0
1570 L$="12 : Y = A+B*ln(X)"
1580 FOR Z=1 TO N
1590 L=X(Z):M=Y(Z)
1600 IF L<=0 THEN R12=0:GOTO 1680
1610 IF L>0 GOTO 1620
1620 C=LOG(X(Z)):D=Y(Z)
1630 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
1640 NEXT Z
1650 R12=1
1660 GOSUB 2010
1670 R43=A:R44=B
1680 F=0:G=0:H=0:J=0:K=0:L=0:M=0
1690 L$="13 : Y = A*exp(B*X)"
1700 FOR Z=1 TO N
1710 L=X(Z):M=Y(Z)
1720 IF M<=0 THEN R13=0:GOTO 1800
1730 IF M>0 GOTO 1740
1740 C=X(Z):D=LOG(Y(Z))
1750 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
```

```
1760 NEXT Z
1770 R13=1
1780 GOSUB 2080
1790 R45=A:R46=B
1800 F=0:G=0:H=0:J=0:K=0:L=0:M=0
1810 L$="14 : Y = A*exp(B*X^2)"
1820 FOR Z=1 TO N
1830 L=X(Z):M=Y(Z)
1840 IF M<=0 THEN R14=0:GOTO 1920
1850 IF M>0 GOTO 1860
1860 C=X(Z)*X(Z):D=LOG(Y(Z))
1870 F=F+(C*D):G=G+C:H=H+D:J=J+(C*C):K=K+(D*D)
1880 NEXT Z
1890 R14=1
1900 GOSUB 2080
1910 R47=A:R48=B
1920 GOSUB 2830
1930 PRINT "============================================================
="
1940 LPRINT TAB(7) "=======================================================
==========="
1950 PRINT "1 for NEW data , 2 to CHECK VALUES , 3 to reprint LAWS or 4 to goto
MENU."
1960 INPUT "Which type do you require ? ",T:IF (T<1)OR(T>4) GOTO 1960
1970 IF T=1 THEN CLEAR:GOTO 110
1980 IF T=2 GOTO 2160
1990 IF T=3 GOTO 190
2000 IF T=4 GOTO 2150
2010 B=(F-(G*H/N))/(J-(G*G/N))
2020 A=((H/N)-(B*G/N))
2030 R=INT(((F-(G*H/N))^2)/(J-(G*G/N))/(K-(H*H/N))*100000!)/100000!
2040 IF C<>W7 THEN C=0 ELSE C=W7
2050 PRINT L$ TAB(24) R TAB(34) A TAB(50) B TAB(66) C
2060 LPRINT L$ TAB(24) R TAB(34) A TAB(50) B TAB(66) C
2070 RETURN
2080 B=(F-(G*H/N))/(J-(G*G/N))
2090 A=EXP((H/N)-(B*G/N))
2100 R=INT(((F-(G*H/N))^2)/(J-(G*G/N))/(K-(H*H/N))*100000!)/100000!
2110 IF C<>W7 THEN C=0 ELSE C=W7
2120 PRINT L$ TAB(24) R TAB(34) A TAB(50) B TAB(66) C
2130 LPRINT L$ TAB(24) R  TAB(34) A TAB(50) B TAB(66) C
2140 RETURN
2150 PRINT:PRINT "* * * PROGRAM FINISHED * * *":PRINT:LOAD "MENU2",R
2160 PRINT
2170 INPUT "Which LAW NUMBER do you wish to check ? ",P:IF (P<1)OR(P>15) GOTO 21
70
2180 IF P=1 THEN Q=R21:S=R22:O=0:GOTO 2330
2190 IF P=2 THEN Q=R23:S=R24:O=0:GOTO 2340
2200 IF P=3 THEN Q=R25:S=R26:O=0:GOTO 2350
2210 IF P=4 THEN Q=R27:S=R28:O=0:GOTO 2360
2220 IF P=5 THEN Q=R29:S=R30:O=0:GOTO 2370
2230 IF P=6 THEN Q=R31:S=R32:O=0:GOTO 2380
2240 IF P=7 THEN Q=R33:S=R34:O=0:GOTO 2390
2250 IF P=8 THEN Q=R35:S=R36:O=0:GOTO 2400
2260 IF P=9 THEN Q=R37:S=R38:O=0:GOTO 2410
2270 IF P=10 THEN Q=R39:S=R40:O=0:GOTO 2420
2280 IF P=11 THEN Q=R41:S=R42:O=0:GOTO 2430
2290 IF P=12 THEN Q=R43:S=R44:O=0:GOTO 2440
2300 IF P=13 THEN Q=R45:S=R46:O=0:GOTO 2450
2310 IF P=14 THEN Q=R47:S=R48:O=0:GOTO 2460
```

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2320 IF P=15 THEN Q=W5:S=W6:O=W7:GOTO 2470
2330 IF R1=0 GOTO 2480 ELSE GOTO 2490
2340 IF R2=0 GOTO 2480 ELSE GOTO 2490
2350 IF R3=0 GOTO 2480 ELSE GOTO 2490
2360 IF R4=0 GOTO 2480 ELSE GOTO 2490
2370 IF R5=0 GOTO 2480 ELSE GOTO 2490
2380 IF R6=0 GOTO 2480 ELSE GOTO 2490
2390 IF R7=0 GOTO 2480 ELSE GOTO 2490
2400 IF R8=0 GOTO 2480 ELSE GOTO 2490
2410 IF R9=0 GOTO 2480 ELSE GOTO 2490
2420 IF R10=0 GOTO 2480 ELSE GOTO 2490
2430 IF R11=0 GOTO 2480 ELSE GOTO 2490
2440 IF R12=0 GOTO 2480 ELSE GOTO 2490
2450 IF R13=0 GOTO 2480 ELSE GOTO 2490
2460 IF R14=0 GOTO 2480 ELSE GOTO 2490
2470 IF R15=0 GOTO 2480 ELSE GOTO 2490
2480 PRINT "This Law Number not used - RESELECT Law Number.":GOTO 2170
2490 PRINT "Check Values for Law Number ";P
2500 PRINT "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~"
2510 PRINT "Actual X      Actual Y         Calc. Y"
2520 PRINT "--------      --------         -------";PRINT
2530 LPRINT TAB(7) "Check Values for Law Number ";P
2540 LPRINT TAB(7) "~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~";LPRINT
2550 LPRINT TAB(7) "Actual 'X'" TAB(20) "Actual 'Y'" TAB(34) "Calc. 'Y'"
2560 LPRINT TAB(7) "----------" TAB(20) "----------" TAB(34) "---------"
2570 FOR T=1 TO N
2580 IF P=1 THEN V=Q+(S*X(T)):GOTO 2730
2590 IF P=2 THEN V=1/(Q+(S*X(T))):GOTO 2730
2600 IF P=3 THEN V=Q+(S/X(T)):GOTO 2730
2610 IF P=4 THEN V=1/(Q+(S/X(T))):GOTO 2730
2620 IF P=5 THEN V=Q+(S*X(T)*X(T)):GOTO 2730
2630 IF P=6 THEN V=Q+(S/(X(T)*X(T))):GOTO 2730
2640 IF P=7 THEN V=1/(Q+(S/(X(T)*X(T)))):GOTO 2730
2650 IF P=8 THEN V=Q+(S*SQR(X(T))):GOTO 2730
2660 IF P=9 THEN V=Q+(S/SQR(X(T))):GOTO 2730
2670 IF P=10 THEN V=1/(Q+(S/SQR(X(T)))):GOTO 2730
2680 IF P=11 THEN V=Q*(X(T)^S):GOTO 2730
2690 IF P=12 THEN V=Q+(S*LOG(X(T))):GOTO 2730
2700 IF P=13 THEN V=Q*EXP(S*X(T)):GOTO 2730
2710 IF P=14 THEN V=Q*EXP(S*X(T)*X(T)):GOTO 2730
2720 IF P=15 THEN V=Q+(S*X(T))+(O*X(T)*X(T)):GOTO 2730
2730 PRINT X(T),Y(T),V
2740 LPRINT TAB(8) X(T) TAB(22) Y(T) TAB(34) V
2750 NEXT T
2760 PRINT "------------------------------------------------------------"
2770 PRINT "Value'A'= ";Q;" , Value'B'= ";S;" Value'C'= ";O
2780 PRINT "============================================================"
2790 LPRINT TAB(7) "------------------------------------------------------------"
2800 LPRINT TAB(7) "Value'A'= ";Q;" , Value'B'= ";S;" Value'C'= ";O
2810 LPRINT TAB(7) "============================================================"
2820 GOTO 1950
2830 L$="15 : Y = A+B*X+C*X^2"
2840 FOR T=1 TO N
2850 X=X(T):Y=Y(T)
2860 W20=W20+1
2870 W9=W9+X:W10=W10+(X*X):W11=W11+(X*X*X):U=X*X*X:W12=W12+(U*X):W18=W18+Y
2880 W19=W19+(Y*Y):W22=W22+(X*Y):U=X*Y:W24=W24+(U*X)
2890 NEXT T
2900 W0=W20
2910 W8=W0*((W10*W12)-(W11^2))+(2*W9*W10*W11)-(W10^3)
```

```
2920 W8=W8-(W9^2*W12)
2930 IF W8=0 THEN RETURN
2940 W6=W0*((W12*W22)-(W11*W24))
2950 W6=(W6-(W9*W12*W18)+(W10*((W9*W24)+(W11*W18)-(W10*W22))))/W8
2960 W7=W0*((W10*W24)-(W11*W22))+(W9*((W10*W22)+(W11*W18)-(W9*W24)))
2970 W7=(W7-(W10^2*W18))/W8
2980 W5=Y(2)-(W6*X(2))-(W7*X(2)*X(2))
2990 W8=W6*((W0*W22)-(W9*W18))
3000 W8=(W8+(W7*((W0*W24)-(W10*W18))))/((W0*W19)-(W18^2))
3010 R15=1:R=W8:A=W5:B=W6:C=W7:GOTO 2120
```

APPENDIX 5

DESIGN DETAILS

APPENDIX 5.1

MODIFICATIONS TO THE ROTARY SLIDING VANE COMPRESSOR TO FACILITATE POWER MEASUREMENT

The following modifications to the compressor were necessary to facilitate power measurement using strain gauges (see Figure 3.8 for sketch of assembled unit):

1. Modifications to the non drive end cover:
   a) Depth of bearing loaction increased by 1 mm, and a 4.5 mm thick spacer manufactured, to accommodate the use of a second seal cover.
   b) Drill and tap 4 off x M8 holes to suit second seal cover (45^{o} to existing holes used for obsolete end plate).

2. Modifications to the drive end cover:
   a) Replace Fag cylindrical roller bearing type NU 2307E with type NUP 2307E to accommodate end float in both lateral directions (necessary since standard end plate replaced by a second seal cover).
   b) Modify existing thrust ring to act as bearing retainer by drilling and tapping 2 off x M6 holes diametrically opposite each other.

3. Modifications to shaft:
   a) Amend shaft register as per 1.a) above.
   b) Dimple shaft to accommodate 2.a) above.
   c) Extend shaft through second seal cover to pick up

slip ring assembly.

d) Eliminate existing cross porting to prevent oil contamination of strain gauge leads.

e) Cross port shaft between first seal cover and engine coupling to allow strain gauge leads to pass through centre of shaft.

4. Modifications to second seal cover (non drive end):

a) Existing cross port to be blanked off.

b) Use existing plugged hole in seal cover to feed oil to non drive end of compressor.

The modified oil feed to the compressor is via two equi-resistance pipes from the oil separator delivering oil to both seal covers.

Engineering drawings showing these modifications are available at the sponsoring organisation.

APPENDIX 5.2

PRINCIPLE OF OPERATION OF THE HEAT PUMP TEST FACILITY

Consider Figure 8.4. Three modes of operation are possible.

a) For use with A.G.R. compressor

gate valve (1) is closed

gate valves (2) and (3) are open.

The majority of refrigerant flows through valve (3), with a proportion being diverted through desuperheater 2 (where it is condensed) and valve (2). The quantity being controlled by the liquid injection valve.

The refrigerant flowing through valve (3) is expanded to the requisite suction pressure, and desuperheated.

b) For use with OS 7061 Bitzer compressor

gate valves (2) and (3) are closed

gate valve (1) is open.

All refrigerant passes through desuperheater 1. The refrigerant gas is then expanded to the requisite suction pressure and returned to the compressor.

c) For use with intermediate units

gate valves (1) and (2) are closed

gate valve (3) is open.

APPENDIX 5.3

DESIGN DETAILS OF EVAPORATER FITTED TO THE PROTOTYPE GAS ENGINE HEAT PUMP

Extract from quotation number 11567/W S. and P. Coil Products Limited [119].

Item	1. DX Coil
Air Volume	11000 cfm
On Temp	$30^{o}C$
Off Temp	$18.32^{o}F$
Load	140,000 btu/hr
Medium	R22
Temperature	$6.5^{o}F$
Pressure Drop	1.37 psi
Coil Code	5 DEH 8.4-26tx58
Face Velocity	658 fpm
Air Pressure Drop	0.45 in wg
Duct Size	41.5 in x 58 in x 7 in
Tube O.D.	15.875 mm
Tube I.D.	15 mm
Equivalent Fin O.D.	25 mm
Fin Spacing	3.2 mm (8 fins per inch)
Fin Thickness	0.2 mm

APPENDIX 5.4

GAS ENGINE HEAT PUMP CONTROL SEQUENCE: SWIMMING POOL APPLICATION

HEATING REQUIREMENT	VALVE A OPEN		VALVE B OPEN		VALVE C OPEN	
	1 - 2	1 - 3	1 - 2	1 - 3	1 - 2	1 - 3
Pool Air Only	X			X		X
Pool Water Only		X		X		X
Pool Air And Water		X		X	X	
Pool Air Only - Discharge Pressure Excessive	X		X			X

Table A5.5.1 G.E.H.P. Control Sequence Swimming Pool Application

APPENDIX 6

SYSTEM RELIABILITY

Table A6.1 Tabulated Breakdowns Experienced On Prototype And Production Gas Engine Heat Pumps

FAULT	CAUSE/REMEDIAL ACTION	NUMBER OF FAILURES
1. Compressor Faults:		
a) Seized Rotor	Seizure occurred after 28 hours operation, and was changed under warranty by the supplier - no feedback was received.	1
b) Broken Vanes and Rotor	Occurred whilst attempting to start compressor against a total hydraulic lock. Manufacturers stated that this should not create a problem, hence a second attempt was made. It was concluded that the manufacturers' statement was incorrect. However, tests have shown that the compressor will pump liquid once it is running.	2
c) Damaged Oil Seals	Excessive defrosting time resulted in a loss of pressure differential between suction and discharge. Oil return to the compressor was therefore prohibited. Remedial Action: Defrosting time limited to a maximum of one minute.	2
d) Collapsed Suction Filter	A collapsed suction filter was drawn into the compression chamber damaging vanes. Remedial Action: Manufacturer supplied reinforced filter.	1
2. Engine Faults:		
a) Excessive Oil Usage	Excessive oil (1 litre per hour plus) used by engine after 1799 hours operation. Engine replaced by manufacturer, under warranty - no feedback was received.	1

	b) Oil Fumes From Rocker Cover	Due to piston ring blow-past exhaust gases can pass into the engine via the piston rings. These gases are normally returned to the inlet manifold, but the return valve was blocked. The gas mixed with some of the engine oil, and was discharged from the oil filler cap as "an oil mist". Remedial Action: Cleaning the return valve added to maintenance schedule. (This problem not experienced on automotive applications).	1
	c) Cylinder Head Failure	Cylinder head replaced by manufacturer under warranty. It was suspected that stellite valves or induction hardening of the valve seats had been omitted.	2
	d) Seized Engine	Incorrect oil added to engine sump during routine service.	1
3.	Serck Heat Recovery Equipment	Excessive corrosion and leaking occurred after less than 1000 hours operation. Remedial Action: Serck heat recovery equipment replaced with Bowman units.	4
4.	Bowman Heat Recovery Equipment	Exhaust gases cooled below dewpoint resulting in increased corrosion and subsequent failure. Remedial Action: Cooling exhaust below dewpoint no longer attempted.	1
5.	Refrigerant Plant Heat Exchangers:		
	a) Incorrect Sizing For Summer Conditions	Excessive discharge temperatures experienced during summer conditions for swimming pool applications, resulting in incidental pressure trips. Remedial Action: Control system modified and firmer guidelines supplied to contracting division.	1
	b) Fin Coil Heat Exchangers Blocked	Ductwork filters fitted as standard on all installations.	1

6. Ancillary Equipment:

a) Oil Filter	Premature failure: cracked casing replaced by supplier under warranty.	1
b) Heinzmann Governor	Flimsy terminals fitted by supplier. Remedial Action: Batch of 6 units returned to supplier for modification.	6
c) Refrigerant Leaks	Damage due to engine/ compressor vibrations. Remedial Action: Improved engine/ compressor mountings.	2
d) Alternator	Alternator design life equivalent to 30,000 to 60,000 miles resulted in repeated failure. Each alternator failure caused the starter motor windings to burn out, as no signal was generated to indicate engine start. Remedial Action: Mains electricity start system adopted.	6
e) Electronic	Luminition electronic ignition failed periodically for no apparent reason. Remedial Action: Replaced by equivalent Lucas systems with no subsequent failures to date.	10
f) Battery	Difficulty experienced when attempting to start engine on cold mornings against loaded compressor. Remedial Action: Mains electricity start system adopted.	3
g) Spark Plugs	Spark plug tracking. Remedial Action: Plugs replaced.	1
h) Oil Pump	The pump feeding oil from the bulk oil tank to the engine sump failed repeatedly after 1000 hours operation. Remedial Action: Sump and bulk oil tank heaters fitted to ensure oil not excessively thickened during periods of shut down.	4